A DESIGN GUIDE FOR NAVAL
SHIP PROPULSION PLANTS

William Robert Michell

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by

WILLIAM ROBERT MICHELL

Submitted to the Department of Ocean Engineering on April 27, 1978, in partial fulfillment of the requirements for the degrees of Ocean Engineer and Master of Science in Naval Architecture and Marine Engineer.

ABSTRACT

A design guide for Naval Ship propulsion plants is developed and a sample problem is presented to demonstrate its usefulness and validity. Although the methodology discussed is not limited in scope, the plants addressed (and for which supporting data is included) are the conventional state-of-the-art plants in use today: Nuclear, 1200 PSI steam, Gas Turbine, Diesel and Combined Plants.

The design approach is preceeded by an introduction to the various propulsion plant functional areas. This discussion centers on the part these functional areas play in the overall design process and on how different plant types have both desireable and undesireable characteristics.

The design approach coupled with the appendices is intended to enable a feasibility study to be made solely from the information contained in this thesis and a reasonable amount of input information. A sample problem using only the design approach and information in the thesis is presented to test this premise.

Thesis Supervisor: Franklin F. Alvarez

Title: Associate Professor of Ocean Engineering

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B.S., Eng. Phys., University of Colorado
(1971)

SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS FOR THE
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and

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LIST OF SYMBOLS

B	Beam
C_p	Prismatic Coeffecient
C_x	Mid Ship Coeffecient
CODAG	Combined Diesel and Gas Turbine
COGAG	Combined Gas Turbine and Gas Turbine
COGAS	Combined Gas Turbine and Steam
COGOG	Combined Gas Turbine or Gas Turbine
COSAG	Combined Steam and Gas Turbine
CRP	Controllable Reversible Pitch
d	Gear Diameter
EAR	Expanded Area Ratio
EHP	Effective Horsepower
EHP_{APP}	Effective Horsepower (with appendages)
EHP_{BH}	Effective Horsepower (bare-hull)
EHP_x	Effective Horsepower (series x)
EOOW	Engineering Officer of the Watch
FOM	Figure of Merit
GT	Gas Turbine
$HP_{SSTG'S}$	Horsepower of Ship's Service Turbine Generator
J	Advance Ratio
K_t	Thrust Coeffecient
KW	Kilowatt

KW_I	Installed Kilowatts
L	Ship's Length
LCC	Life Cycle Cost
MDCS	Maintenance Data Collection System
MDT	Mean Downtime
MTBF	Mean Time Between Failure
MTBM	Mean Time Between Maintenance
MTTR	Mean Time To Repair
ND	Naval Distillate
NSFO	Navy Special Fuel Oil
P/D	Propeller Pitch to Diameter Ratio
PC	Propulsive Coeffecient
PSI	Pounds per Square Inch
q_t	Dynamic Pressure
R	Resistance
R_{APP}	Resistance With Appendages
RMA	Reliability, Maintainability and Availability
RPM	Revolutions Per Minute
R_T	Towed Resistance
R(t)	Reliability (as a function of time)
SFC	Specific Fuel Consumption
SHP	Shaft Horsepower
SHP_I	Installed Shaft Horsepower
T	Draft
T	Thrust
t	Thrust Deduction Factor

V	Ship's Speed
V_a	Speed of Advance
V_{end}	Endurance Speed
V_{sus}	Sustained Speed
W	Weight
w	Wake Fraction
W_f	Fuel Weight
W_{200}	Weight Group 200
Δ	Displacement
η_H	Hull Effeciency
η_o	Open Water Propeller Effeciency
η_R	Relative Rotative Effeciency
ρ	Density of Salt Water
σ	Local Cavitation Number
λ	$\frac{1}{MTBF}$
μ	$\frac{1}{MTTR}$

INTRODUCTION

Naval ships are designed for payload considerations, not propulsion plants; yet, when considering such factors as weight, volume, manning and cost the propulsion plant normally has the most significant impact on the ship, and thus upon payload. Typical ranges of influence in these areas are (1)[1]:

DESIGN CHARACTERISTIC	PERCENT OF TOTAL SHIP
WEIGHT (including fuel)	30%-40%
VOLUME (including fuel)	20%-30%
COST: acquisition	10%-25%
life cycle	20%-30%
MANNING (engineering)	20%-25%

In addition to the above characteristics there is considerable impact on Logistic Support. Propulsion plant maintenance represents a large percentage of the work load during major overhauls and at the intermediate support level. Regardless of how you view ship design, it would be hard to find a functional design area with more impact than propulsion.

Figure 1 shows how propulsion fits into the overall design process (1).

Beyond the obvious propulsion design objectives of optimizing cost and performance are still more specific goals.

1 Numbers in parenthesis refer to References

SHIP SYSTEM

CONTAINMENT	MISSION SUPPORT	MOBILITY	SHIP SUPPORT	HUMAN SUPPORT
hull structure	armament	* PROPULSION	pollution control	habitability
hull form	command & control	maneuvering	fluid & gas sys	hotel services
mass properties	exterior comm	navigation	ECS	
ship arrgmts	surveillance	anchoring mooring & towing	materials handling	
	counter measures		interior comm	
	fire control		electric plant	

FIGURE 1

The relative importance of these goals will depend primarily on the requirements, constraints and philosophies generated in the early stages of conceptual design.

DESIRED PROPULSION SYSTEM DESIGN GOALS (1)

HIGH	LOW
power	volume
availability	weight
reliability	fuel consumption
	cost
	maintenance
	manning
	noise
	risk

Even the best propulsion designer would find it impossible to optimize all the above characteristics in his design; this is not even a rational objective. What the designer does attempt to do is tailor the design to best satisfy the established requirements for a particular ship. Arriving at a 'best' propulsion plant in this manner will depend on the soundness of the design methodology.

This thesis presents a design methodology which enables the propulsion plant designer to select a propulsion plant given the design requirements, constraints and philosophy. It is limited to the conceptual design phase, and therefore the end results may not be the 'best candidate' but the 'best candidates'. However, since propulsion plant design

is an iterative process the methodology need only be re-applied, with increasing levels of detail, to arrive at a single plant.

Section 1 is an introduction and/or review of the various functional areas that comprise the propulsion system. Each area is discussed with respect to its contribution to the system as a whole. Also, within each area, the different components available for use are addressed in some detail. Their advantages, disadvantages and impact on subsequent functional areas are presented in this section.

Section 2 introduces a design methodology, which can be used in selecting a 'best' propulsion plant for a given ship: From among those analyzed. The approach outlined permits the designer to determine the adequacy of various components available in each functional area:

ENERGY PRODUCTION
ENERGY CONVERSION
POWER GENERATION
POWER TRANSMISSION

Explaining the best way of integrating the different components into a feasible plant and sizing the finished product is a major objective of this section.

Section 3 is an example propulsion plant design. It employs the methodology presented in Section 2; and is

supported by information contained in the thesis. The example starts with basic inputs, not unlike the information available in a real design situation. The end result of the example is a propulsion plant capable of meeting all the desired requirements; and one that is a best fit, with respect to the design inputs.

The Appendicies go beyond the normal contribution of just supplemental information. They are, in fact, the key to making this thesis a self-contained conceptual propulsion plant design workbook. An in-depth discussion, and accompanying example, of each major design step can be found in the appendicies. There is also a complete set of weight and volume graphs covering almost every major propulsion plant component. This is similar to information found in most good synthesis design programs, but in a more useable format. Also included is performance data on specific propulsion plant machinery.

1. PROPULSION PLANT FUNCTIONAL AREAS

1.1 INTRODUCTION Propulsion plant design is a process that combines energy converters, prime movers, transmissions and propulsors in a variety of ways to obtain a number of candidate plants. The overall system performance, size and weight of these synthesized models will ultimately determine which plant is to be selected. And since system characteristics are a consequence of the individual components, the advantages, disadvantages and operating peculiarities of these components will be discussed first. The discussion assumes a basic familiarity, on the part of the reader, with the fundamental operating theory of each subsystem element. Table 1.1 lists propulsion components according to their functional area.

1.2 ENERGY SOURCES The two basic types of fuel commonly stocked at bunkering stations for marine use are the heavy residuals and the more refined distillates. When Navy Special Fuel Oil (NSFO) was the accepted fuel for most Navy ships (steam plants), the economics of burning distillates could have had an influence on propulsion plant selection. Refining costs make distillate grades more expensive than NSFO. Specifically, plants capable of burning the cheap,

PROPULSION PLANT FUNCTIONAL AREAS

ENERGY SOURCES

1. Marine Diesel
2. JP-5
3. NSFO
4. Nuclear Fuel
5. Naval Distillate (ND)
6. Bunker-C

ENERGY CONVERSION DEVICES

1. Oil Fired Steam Generator (Boiler)
2. Diesel Engines (Internal Combustion)
3. Gas Turbine (Internal Combustion)
4. Nuclear Reactor

POWER GENERATORS

1. Steam Turbines
2. Gas Turbines
3. Diesel Engine

POWER TRANSMISSIONS

1. Speed Reduction: Reduction Gears
2. Power Transmission: Shafting and Bearings
3. Thrust Production: Propellers
    1. fixed pitch
    2. controllable pitch
    3. jet pump
4. Electrical

CONTROL

1. Manual
2. Semi-Automatic
3. Automatic

Table 1.1

heavy residuals would have a lower life-cycle cost (LCC). But this factor no longer influences propulsion plant selection.

In an effort to reduce maintenance and enhance the reliability/availability of 1200 psi boilers, the Navy converted all of its steam plants to Navy Distillate (ND) in the 1970's. The results to date suggest the tradeoff was indeed cost effective. Since the deisels used in Navy ships burn either marine diesel or JP-5, ND was eventually dropped completely. The Navy now uses only marine diesel and JP-5: a good move considering the increase in gas turbine plants. Gas turbines are more sensitive to fuel impurities than are boilers, and would have required the distillate fuel regardless of what the steam plants were using.

Although cost differentials between grades of fuel oil will no longer have a significant effect on Navy propulsion plant selection, nuclear fuels vs fossil fuels will still impact. Nuclear proponents argue that the reduced LCC of their fuel, coupled with the increasing prices of conventional fuels, weighs in their favor. The opponents counter with increased initial costs, manpower training costs and the political (and social) issues that continually shroud nuclear power. But the emphasis on these arguements vary from year to year; making the energy source costs an unpredictable impact on propulsion plant selection.

1.3 <u>ENERGY CONVERSION/POWER GENERATION</u> Since energy conversion devices and power generators are either integral parts (gas turbines and diesels) or almost always considered as one (boilers and steam turbines), they are discussed as systems rather than individual components.

1.3.1 <u>GAS TURBINES</u> The gas turbine engine combines energy conversion and power generation into a single unit. The two basic sections are a gas generator and a gas turbine. Intake air is compressed and burned in the generator and then expanded through the turbine. Figure 1.1 shows a simple GT with the internal, as well as functional, relationships. The reason for compressing the air prior to combustion, is that gases at atmospheric pressures do not provide enough energy to produce useful torques upon expansion.

The GT requires six to ten times the amount of air of a comparably rated reciprocal engine. This requirement, when translated into intake and exhausting plenums, decreases the power density (SHP/ft^3) considerably. In fact, GT plants impact heavily on topside arrangements for this same reason. The GT is also sensitive to the purity and amount of turbulance of combustion air, which makes intake engineering critical.

As much as 73% of the energy produced in a simple GT can

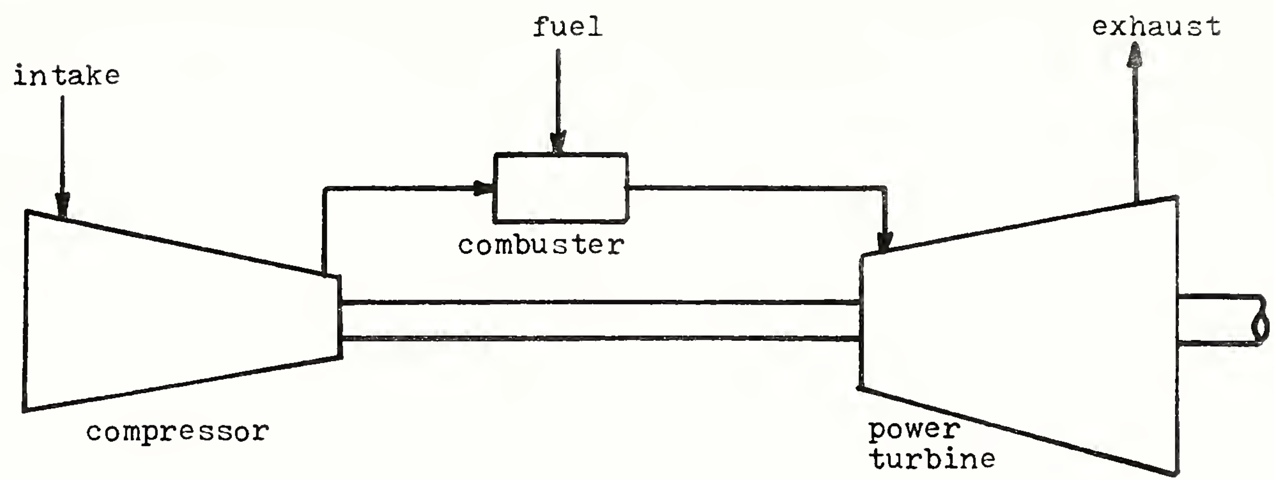


FIGURE 1.1 SIMPLE GAS TURBINE

be lost in the exhaust; therefore, GT's are often used in conjunction with some energy recovering device (i.e. waste heat boiler). Furthermore, the GT exhibits poor off-power specific fuel consumption (SFC). Other disadvantages of the GT propulsion plant are: Personnel hazards from airborne noise, SFC sensitivity to air temperatures and unidirectional operation.

The unidirectional operation necessitates the use of some reversing device in power transmission. Whether it is in the form of clutches, reversing reduction gears or reversible pitch propellers, the impact is both technically difficult and expensive. To date the Navy has only used clutches and reversible pitch propellers, but research continues in the other areas.

There are several significant advantages of GT plants which have resulted in their increased use in Naval Combatants. Probably the most important of these is their low specific weight (lbs/SHP) and compactness of the prime mover. These characteristics result in minimal installation efforts and ease of removal as entire units. Quick response, fast start ups and ease of automation are three other GT traits desireable for Navy use.

Because Naval application of GT's has been limited to first and second generation aircraft derivitive models,

their usage has been chiefly in light and medium displacement ships. Table 1.2 lists some gas turbines currently available for marine use.

MODEL	POWER RATING(hp)	MODEL	POWER RATING(hp)
GE LM2500	20,000	FT12A-3	2,500
GE LM1500	12,500	FT12A-6	3,150
GE LM 500	4,560	FT9	
GE LM 100	1,000	501K	3,780
TGP/GTPF 990	5,000	T3001	3,000
FT4A-2	24,200	TF12A	1,000
FT4A-12	26,950	TF14B	1,250
FT4A-14	31,150	TF25A	2,000
FT4C-2	35,500	TF35	2,500

TABLE 1.2 U.S. MADE, LIQUID FUELED MARINE GT'S

1.3.2 STEAM BOILERS AND TURBINES There is no existing propulsion system in Naval use today with service experience approaching that of the steam-generator (boiler) steam turbine combination. Figure 1.2 is a schematic diagram of a steam plant similar to those found on Navy ships today. These years of steam plant experience have resulted in high system availability, inherent component reliability, readily available parts and experienced manufacturers. It's biggest advantages though are in areas such as cycle flexibility, ability to burn low grade residual fuels, the ease of low speed and reversing operation, and its low temperature thermodynamic cycle.

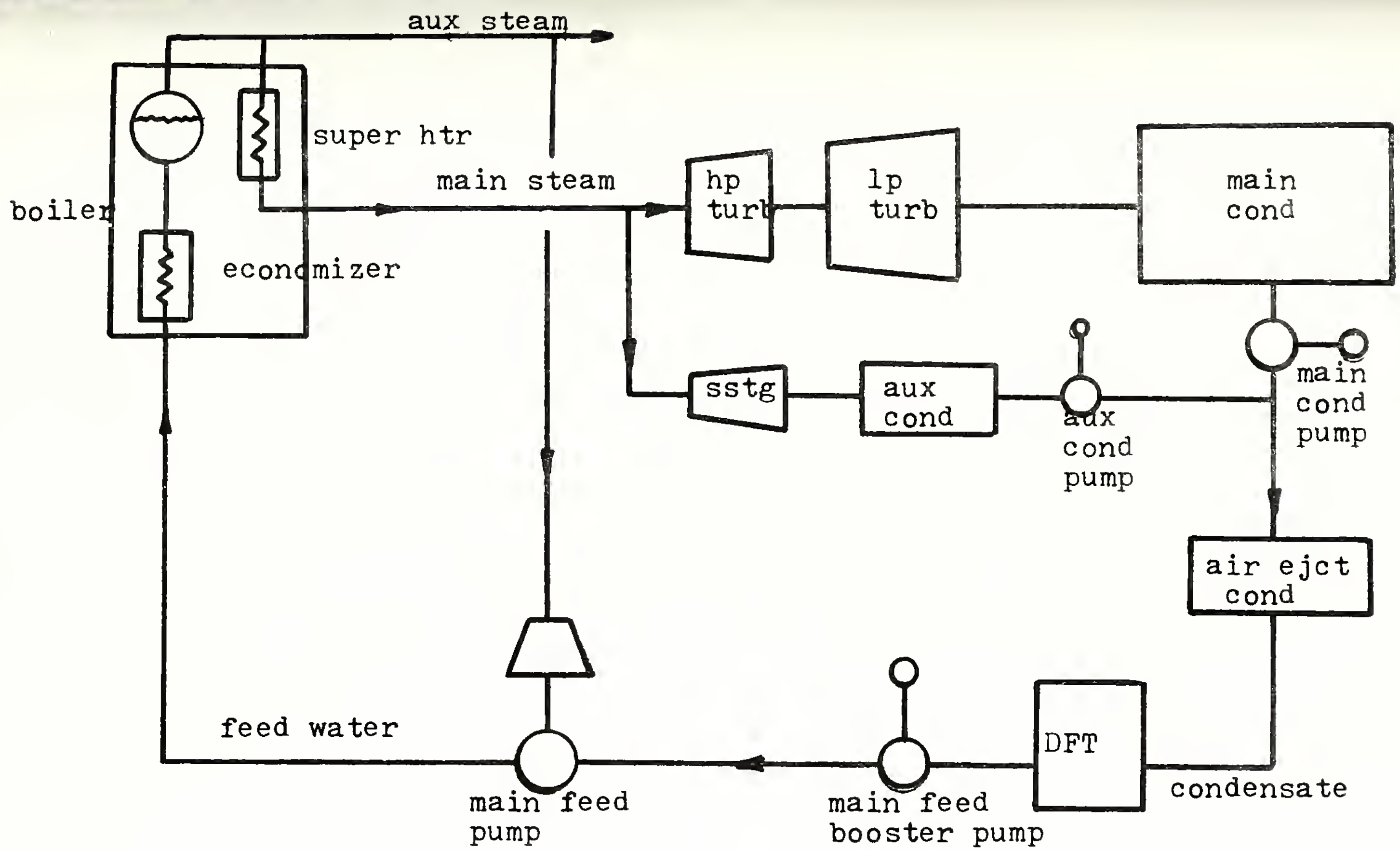


FIGURE 1.2 STEAM PLANT SCHEMATIC

Like all propulsion systems steam also has its drawbacks. Unfortunately, for steam, those less desireable attributes are in areas of major concern in today's Navy: Low power density and high manning requirements. Slow start-ups and a large number of supporting subsystems are additional disadvantages of the steam plant.

In the 1950's steam plants went to higher pressures (1200 psi) in an attempt to increase SHP without an appreciable increase in machinery size. Lately, because of increased maintainance in 1200 psi plants, there is talk of reducing pressures to near 800 psi.

Because of the lack of high powered GT's and the weight of high powered diesels, steam plants (along with nuclear power) are considered superior in high SHP applications: Greater than 80,000 HP.

1.3.3 DIESEL ENGINES When the entire marine industry (commercial, recreational and Naval) is taken into consideration, the diesel engine is the most widely used prime mover. This popularity is, in part, linked to the fact that there are several classes of diesels allowing it to cover the complete spectrum of power ranges:

CLASS	RPM	POWER RANGE	APPLICATION
High Speed	1800-3000	Up to 500 SHP	Generator Drive Small Boats
Medium High Speed	720-1200	500-3600 SHP	Ferrys Tugs
Medium Speed	400-500	5000-12000SHP	Used singly or in tandum for powers up to 24 K.
Slow Speed	110-130	4000 HP per cylinder	Normally found in ships requiring 20K - 40K SHP

The most attractive features of the diesel engine plants from a commercial standpoint are its low SFC and ability to burn low grade fuels without an appreciable increase in maintenance. But for Naval application, low manning, ease of automation, fast start-ups and quick response are its more important characteristics.

The reason diesel engines have not found better acceptance in Naval combatants is their low power to weight ratio. Large commercial ships and Naval auxiliaries (with their slow speed requirements and high displacements) can live with this, but it has all but eliminated diesels in high speed combatants. Other disadvantages are their high self-generated noise levels, poor slow speed operations and the necessity to clutch or use a CRP for reversing operations.

1.3.4 NUCLEAR POWER No propulsion plant has generated as much political and social controversy as nuclear power. The potential radiation hazard associated with nuclear plants have kept them in the public lime light, but nuclear power's record over the past twenty years should convince the most skeptic opponent of its safety. Politically (with respect to its use in Naval ships) the controversy centers around nuclear power's initial cost, which is extremely high. This, of course, is offset by the reduced LCC, so say nuclear proponents. Beyond these issues lay even more fundamental advantages and disadvantages, from a design viewpoint.

Nuclear power is used in conjunction with steam turbines but unlike boilers can offer almost instant response to changes in steam demand. Nuclear power also requires no air and generates no exhaust, thus eliminating the need for uptakes. This fact accounts for a low power density and delights combat systems designers, who are always searching for more freedom in topside arrangement.

Undoubtedly the best feature of nuclear power propulsion for military use is its unlimited range.

The two main disadvantages of nuclear power are a result of the radiation produced during operation; first it is a potential health hazard and secondly the shielding necessary to reduce its effects result in a low power to weight ratio.

1.4 COMBINED CYCLE PLANTS In an attempt to capitalize on the advantages offered by various power generating systems, combinations of prime movers are often utilized in propulsion plant design. An example of this would be the CODAG plant (COmbined Diesel And Gas Turbine). In this arrangement a small medium-high speed diesel is used for normal cruising or endurance speeds giving good SFC's and reducing endurance fuel weights; while a gas turbine is installed to assist in providing for high speed operations. Another example might be a COGOG plant (COmbined Gas Turbine Or Gas Turbine), where a large and small GT are coupled to the same shaft; the smaller turbine providing power up through the cruising ranges and the larger turbine handling high power requirements.

Whenever combinations such as these are used, particular attention must be paid to the interfacing problems. While enhancing the separate advantages and possibly cancelling the less desireable effects of each system, the designer must be on guard for additional disadvantages arising from their combined use: Such as clutching problems associated with GT's and diesels mechanically linked to the same reduction gears.

The single biggest reason for using combined cycle plants is increased plant efficiency. High speed requirements of Naval combatants dictate high installed SHP, even

though these speeds are called upon less than 10% of the time. This results in the propulsion plants operating at less efficient off-design power levels most of the time. The objective in improving plant efficiency is not the savings in fuel dollars, but the savings in fuel weight. Reduction in endurance fuel weight, due to improved 'overall SFC's', can be transformed into payload: A very desireable tradeoff in Navy combatants.

Combined cycle plants also provide a means of increasing the horsepower output of a prime mover above its base rating. This approach can be seen in a COGAS plant (COmbined Gas Turbine And Steam). In this setup the GT exhaust is directed into a waste heat boiler, thus producing steam. The steam then drives a steam turbine that in turn assists the GT. This particular combination could result in a GT rated at 25,000 hp actually being the primary energy source for a 35,000 hp propulsion plant.

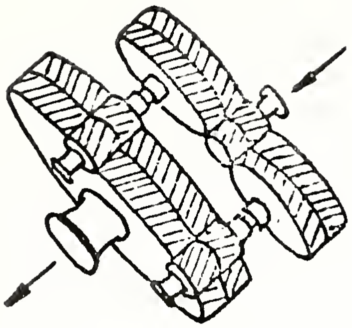
1.5 POWER TRANSMISSIONS Although electrical power transmission can be found in marine use (most common is submarines), it is not employed in Naval surface ship propulsion at this time. And for this reason it is not addressed in this thesis. It should be pointed out, however, that present reasearch in super conducting machinery may make electric propulsion a viable alternative in the future.

1.5.1 <u>REDUCTION GEARS</u> Except for low-speed diesels and electric drives, all prime movers are coupled to propulsors through reduction gears. This allows both prime mover and propulsor to operate at their most effecient RPM. Figure 1.3 shows the most common types of marine reduction gears.

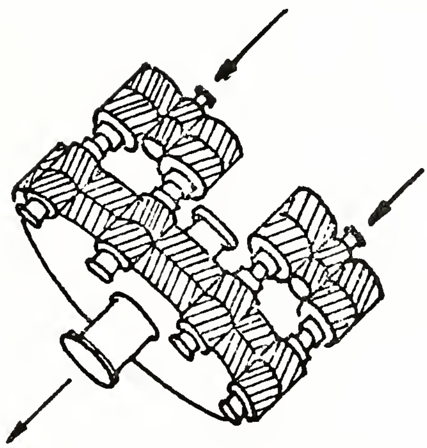
Because weight and size are always a factor in propulsion design, the double reduction, double input, locked train reducer is standard for most high power marine requirements. The locked train arrangement splits the torque input between two gears, reducing the size of the first reduction gears: The two smaller gears have less weight than one larger one. This weight reduction offsets the added parts and the need for torsionally flexible connections between first reduction gears and second reduction pinions.

Planetary gears offer the least weight and smallest envelope for a particular set of conditions. Even with this advantage over standard gears, the Navy has not yet moved in the direction of epicyclic reduction gears. The technical difficulties such as ring gear flexibility, planet gear load-sharing capabilities, planet gear lubrication, severe tolerances and bearing design have had much to do with the Navy's reluctance to use them.

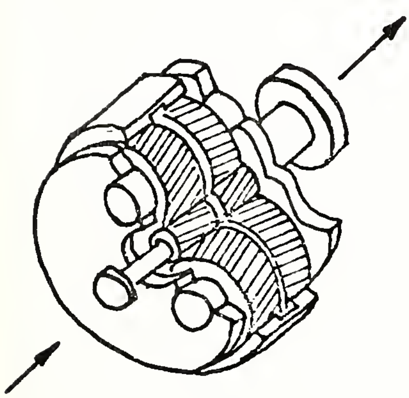
Like all system components, reduction gear design is left to the experts, but it is necessary for the propulsion plant designer to be able to size them. Appendix III shows



DOUBLE-REDUCTION
SINGLE-INPUT
LOCKED-TRAIN
REDUCER



DOUBLE-REDUCTION
DOUBLE-INPUT
LOCKED-TRAIN
REDUCER



SINGLE-REDUCTION
PLANETARY
REDUCER

FIGURE 1.3

how this is accomplished.

1.5.2 <u>SHAFTING</u> The shafting (including shaft bearings) has three primary functions.

1. Transmit power from prime mover to propulsor
2. Support propulsor
3. Transmit thrust from propulsor to ship's hull.

The design requirement in determining shafting material and dimensions is, quite simply, to meet the above objectives with a minimum impact on weight.

Shafting should be able to handle expected torques, bending moments and thrusts with optimum dimensions and bearing spacings. Of these loads, torsion and bending normally have the greatest impact. The tosional load has the following relationship to SHP and RPM:

$$\text{Torsional Load} \propto \frac{SHP}{RPM}$$

The Navy usually increases the expected torsional load by factor of 20% as a margin of safety. The most significant bending load is due to the propeller:

$$M_p = W_p L_p$$

M_p - moment due to propeller

W_p - propeller weight (in salt water)

L_p - length between propeller center of gravity and the reaction point of the last shaft bearing

Another shafting tradeoff is hollow vs solid: CRP's need hollow shafts for hydraulic lines. The smaller the shaft diameter (solid shaft) the smaller the strut bearings; which reduces appendage drag. The hollow shaft, on the other hand, reduces shaft weight. The degree of weight reduction depends on the ratio of the inside and outside diameters. The shear stress in a hollow shaft is related to the inside and outside diameters in the following manner:

$$S \propto \frac{1}{D_o^4 - D_i^4}$$

So D_i/D_o also becomes a tradeoff factor. Decreasing this ratio reduces the shaft weight but also reduces the shear stress margin. All of the factors mentioned here are addressed in the shafting design process. References 14 and 15 contain this and additional information for the shaft design. However, for the scope of this thesis only shafting weight is of significance and this information is found in Appendix IV.

1.5.3 PROPELLERS Propeller effeciency is the most dominent factor in determining the propulsive coeffecient. It is therefore necessary to make a preliminary propeller selection in order to determine the required SHP. Selection depends on the number of shafts and the type of plant:Gas turbines and some diesel plants require reversible pitch propellers. Sel- criteria varies but the following axioms usually hold:

1. Maximum effeciency at endurance speeds.
2. Minimum cavitation at maximum sustained speeds.

In general higher efficiencies are associated with large diameters and low RPM's. The diameter of course is limited by the ship's draft and hull clearance: Normally hull clearance should be about 25% of the propeller diameter(1). For a given diameter the designer can construct an efficiency vs RPM curve. Selecting an RPM from this curve is not always easy. For instance, using an RPM slightly higher than the one corresponding to the maximum efficiency may have a positive impact on machinery weight. This is especially evident in diesels where there is a significant weight difference between low speed, medium speed, medium-high speed and high speed units. For steam turbines, gas turbines and diesels, an increase in RPM serves to reduce the size of the reduction gears.

Where Controllable Reversible Pitch propellers (CRP) are needed, the expanded area ratio (EAR) is important. This is because if EAR is greater than about .78, the blades will not have enough clearance to pass by each other when reversing. CRP's are also restricted to 40,000 SHP (presently) because of mechanical difficulties. The advantages of a CRP are:

1. The ships speed can be changed while running the prime mover at its most efficient RPM.
2. Instant reversing.

If, instead of a diameter, an RPM is specified then the propeller analysis concerns itself with finding an optimum diameter. Appendix II shows the entire propeller selection process.

1.6 CONTROL Fully automatic, remote controlled plants are state-of-the-art. Although this thesis does not address the control aspect of propulsion design, beyond indicating the ease with which a plant-type can be automated, its importance does deserve comment.

Automation is relatively inexpensive with respect to weight and volume but does have some minor technical drawbacks. One arguement, which will take time to evaluate, is the maintainability. This will depend not only on the inherent reliability of the components but on the skill level of the engineering department personnel.

Probably the biggest advantage of automation is its affect on manning reductions. This also helps designers working on habitability problems.

1.7 CONCLUSION As can be seen in this section, there are a wide variety of possible propulsion systems a designer can utilize. Each system or combination of systems has its own unique characteristics, which make it the 'best' choice,

depending on the design requirements, constraints and philosophy employed. The final determination of which plant optimizes a design is normally accomplished through a quantitative selection analysis: See Section 2.10.

Appendix I contains a functional schematic of several popular power generation systems, along with their advantages and disadvantages. This information is intended to provide a basic shopping list for possible propulsion plants. It should aide in determining obvious non-candidates and furthermore suggest which systems are the most likely to meet the design requirements.

2. CONCEPTUAL DESIGN PROCESS

2.1 <u>INTRODUCTION</u> The objective of the conceptual design stage (for propulsion plants) is the determination of a feasible and somewhat optimum propulsion plant. Because it is the type of plant, not specific components, that is of major importance, the designer must concern himself primarily with the functional relationships within the plant. These functional areas can be classified as follows:

1. Energy Conversion
2. Power Generation
3. Power Transmission
4. Propulsion

Within each functional area there exists several candidates, the characteristics of which will be inputed into a selection analysis. From this will emerge a 'best' candidate (or candidates).

The entire propulsion plant design process is similar to the initial study except that the level of detail increases in successive stages. Functional relationships are replaced by specific relationships, emperical data is replaced by component data and interfacing plays a more dominant role.

2.2 FUNCTIONAL RELATIONSHIPS Defining characteristics within, and relationships between, the different functional areas is the first step in conceptual design. This provides an overview of the possible combinations that exist, and also quickly demonstrates infeasible combinations. Figure 2.1 shows some interrelationships of different functional area components. It should be evident from this figure that certain component combinations are not practical. For example, a steam turbine would never be used with a clutch where as a diesel may or may not use one.

Awareness of the functional relationships insures that the designer knows the subsequent interfacing consequences of a particular component selection.

2.3 PRIME MOVERS If any propulsion plant component could be considered the focal point, it would be the prime mover. It is the prime mover that usually determines which propulsor and drive train to use; and it also determines which energy conversion means is to be utilized. Because of their importance it is normal to classify propulsion plants according to their prime movers (except for nuclear power). This can be seen in more detail in Appendix I.

Another reason prime movers are usually the primary concern in propulsion plant design is that they are a controlling factor for weight and volume. They either directly

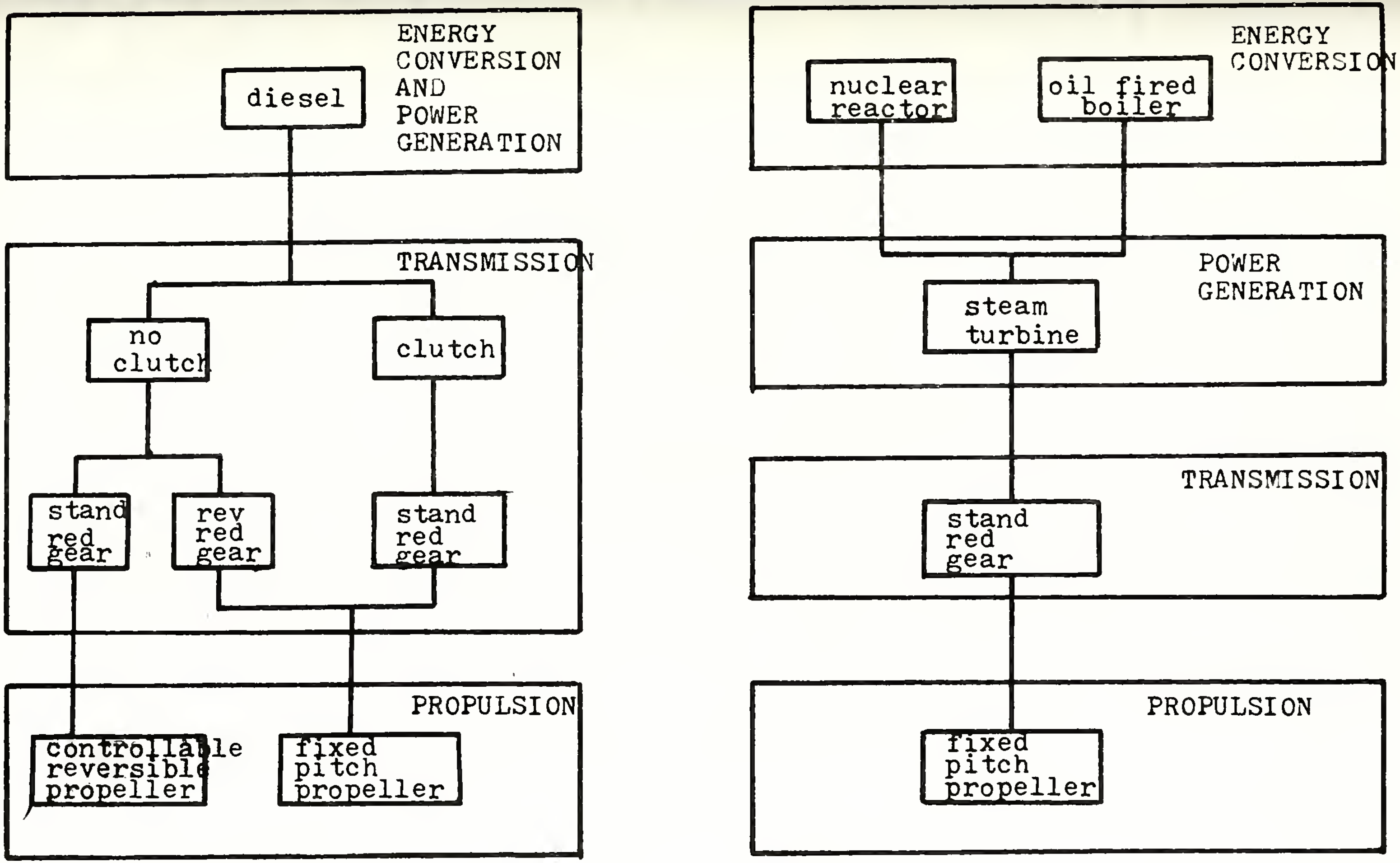


FIGURE 2.1 FEASIBLE PROPULSION PLANT FUNCTIONAL COMBINATIONS

or indirectly dictate plant size.

2.4 DESIGN INPUTS Like all designs the starting point for a propulsion plant design begins with the inputs.[1] Some quantities are fixed (endurance speed, maximum sustained speed, etc.), some are just estimates (full load displacement) and others are undefined. It is up to the designer to integrate the quantitative and qualitative aspects of the inputs to insure that the final product is the one that best suits them.

The design requirements, design constraints and design philosophy are the chief source of the design inputs. Such things as mission, maximum displacement, maximum sustained speed, endurance speed, etc. are usually defined in these sources. Quite often a propulsion plant design philosophy does not exist 'per-se', and therefore must be interpreted from the overall ship design philosophy. For instance a desire to minimize displacement is obviously interpreted as a restriction on propulsion plant weight.

The first goal of the propulsion designer is to turn the design inputs into a shaft horse-power requirement: He can then determine the best method for supplying that horse-power.

1. Cost is always an important constraint/requirement in any design, but it will not be discussed here.

2.5 PRE-DESIGN SELECTION Very often the design philosophy, design requirements or design constraints will not permit the use of a specific plant type; for instance its useless to consider state-of-the-art nuclear propulsion for a light weight-high speed frigate because of the weights involved. Therefore, before applying the design methodology to individual plant types, it pays to subject their gross characteristics to a 'go-no-go' study. This saves time by eliminating any obvious non-candidate plants from an in depth quantitative analysis. There may by other plants eliminated later on in the design process for other unforseen reasons, but at least they must be included at the beginning of the analysis phase.

2.6 INSTALLED SHAFT HORSEPOWER Determining installed shaft horsepower (SHP_I) is a four step operation:

1. Find the horse power required to propel the bare hull through the water: EHP_{BH}.
2. Adjust EHP_{BH} to allow for appendage resistance: EHP_{APP}.
3. Calculate the shaft horsepower necessary to overcome all losses and provide the required EHP_{APP}: SHP.
4. Increase the SHP by some design factor to allow for design unknowns: SHP_I.

2.6.1 EFFECTIVE HORSEPOWER (BARE HULL) What is EHP_{BH} and how is it calculated? Basically EHP_{BH} is the power the ship must deliver to the water in order to propel itself. The ship in this instance refers to the basic hull form free from any appendages associated with steering, propulsors, weapons, etc. The magnitude of the resistive forces that must be overcome by EHP_{BH} (friction forces, wave-making forces, etc.) vary at different speeds; making a speed vs resistance graph a good yardstick for the range of EHP_{BH} required. For the ranges of $V/\sqrt{L}$ that Naval combatants are designed for, wave making resistance dominates; this factor drives the hull form to one of increased length and fineness. And since horsepower varies at greater than the cube of the speed this has a large impact on the size of the propulsion plant.

Model testing or 'series analysis' are two methods used to estimate ship resistance. Although model testing is more exact, it is expensive and not always practical in the early stages of design. Because of the established relationships between hull form and resistance, the series approach is based on previous parent-model tests; and, in early stages of design, it offers the advantages of speed and flexibility. Reference (2) outlines both methods in a step by step example.

If the series approach is used the resultant horsepower

is usually refered to as EHP_x (where x refers to the type series used) and is converted to EHP_{BH} using a worm curve. This is an emperical curve that accounts for the difference between the parent hull forms and most destroyer hull forms.

2.6.2 EFFECTIVE HORSEPOWER (WITH APPENDAGES) Going from EHP_{BH} to EHP_{APP} would be easy if a model were available, but as stated previously this is quite often not the case. Hull projections such as struts, rudders, bilge keels, cathodic protection devices and other common appendages can easily be accounted for through emperical design curves, because their affects are predictable on similar hull forms. Figure 2.2 is an appendage curve applicable to most cruiser/ destroyer hulls (3). It is actually a ratio of resistances, but EHP and resistance are directly proportional.

Although they constitute the greatest deviation from the baseline hull form, sonar domes have little effect on propulsion plant selection. This is because they are normally designed such that they contribute little or no resistance (wave making) at maximum sustained speed; and this, of course, is the same speed that determines propulsion plant size. This is not to imply that the dome does not impact plant size, it can have a pronounced affect on the amount of endurance fuel because of the increase in required horse-

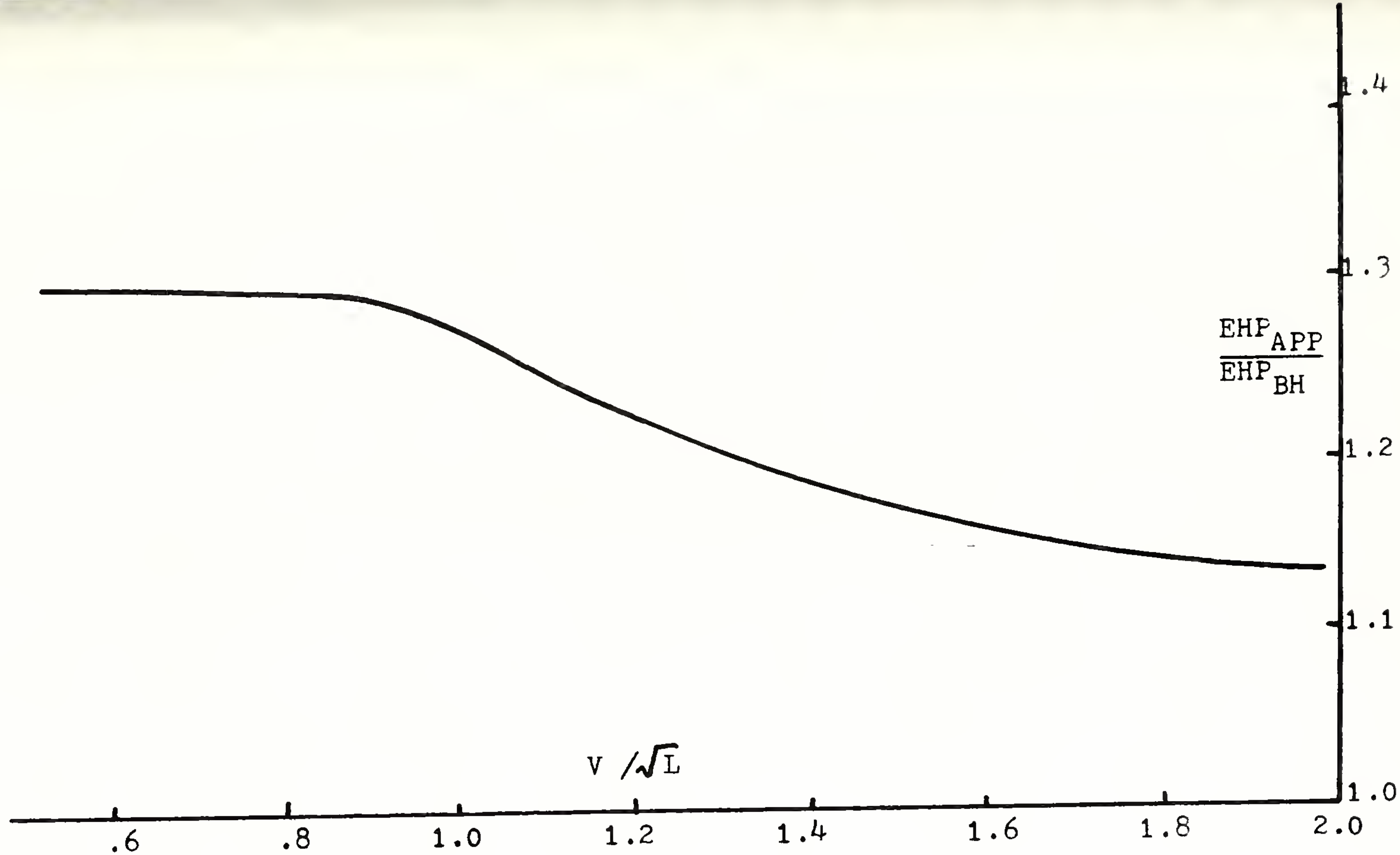


FIGURE 2.2 CRUISER/DESTROYER APPENDAGE CURVE

power at these speeds. The increase will depend on the size relationship between the dome and hull. Obviously the contribution of a sonar dome to the ship's sectional area curve would be much less on a 600 foot cruiser than a 400 foot destroyer; but as a guideline (on the high impact side) Figure 2.3 shows the study of sonar dome affects on the horsepower of a 350 foot destroyer.

2.6.3 <u>SHAFT HORSEPOWER (SHP)</u> After EHP_{APP} is established the designer must determine the propulsive coeffecient (PC): This factor accounts for the losses in going from EHP_{APP} to SHP. Calculating PC begins with the propeller analysis. The importance of the propeller selection is obvious when you consider that it represents 95% of the loss in power going from the prime mover to the water. For most cruisers and destroyers PC varies from 0.60 to 0.70, while the propeller effeciences go from 0.65 to 0.72.

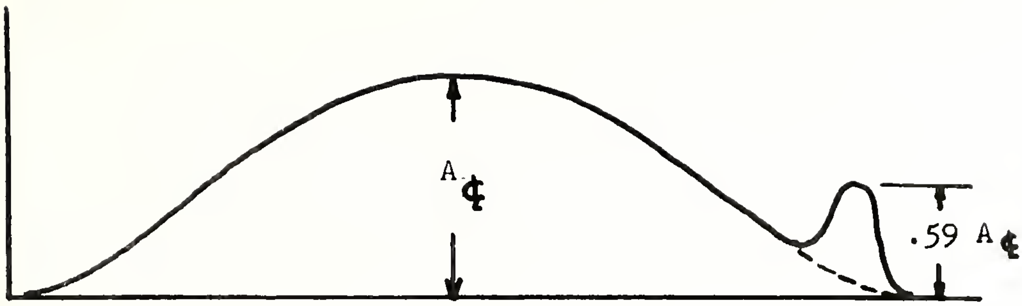
Once the propeller has been found, the other factors affecting PC must be determined. The formula for PC is:

$$PC = \eta_O \cdot \eta_H \cdot \eta_R$$

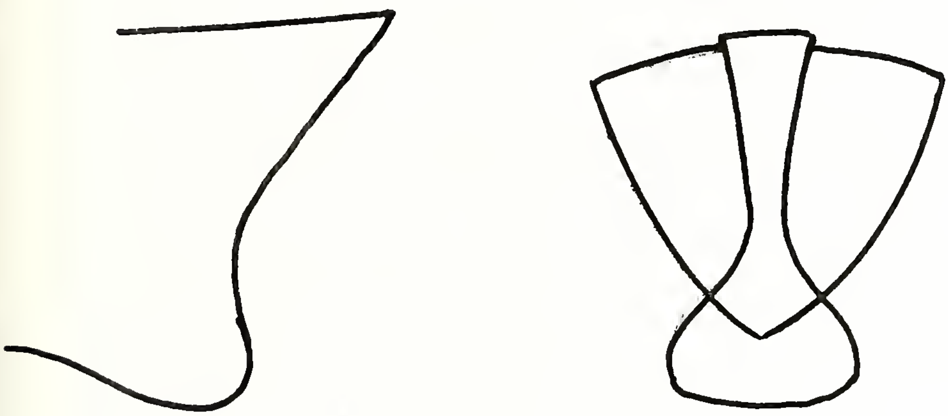
η_O = Open water propeller effeciency

η_H = Hull effeciency

η_R = Relative Rotative effeciency



SECTIONAL AREA CURVE



SMALL 350' DESTROYER

WITHOUT SONAR		WITH SONAR	
Δ	2300 tons	Δ	2458 tons
WETTED SURFACE AREA	22,800 ft^3	WETTED SURFACE AREA	24,190 ft^3

V_{kts}	PERCENT CHANGE IN EHP WITH SONAR DOME
12	102%
16	49.6%
20	27.4%
24	4.2%
28	0.0%

FIGURE 2.3 SONAR DOME AFFECTS ON EHP

η_R seldom varies over two percent (0.98 - 1.02) and can therefore be set equal to one for propulsion plant design purposes. η_H, on the other hand, must be addressed in more detail.

$$\eta_H = \frac{1-t}{1-w}$$

$$w = \frac{V-Va}{V} \text{ ; wake fraction}$$

$$t = 1 - \frac{R_T}{T} ; \quad \text{thrust deduction factor}$$

Wake fraction and thrust deduction factors account for the differences in resistance arising from the ship-propeller interactions. Naval destroyers have wake fraction values between -0.02 and +0.02 for ship with struts and between 0.04 and 0.08 for ships with bossings. The value of t, to a first approximation, may be assumed to be equal to w (2).

The Navy design convention in determining horsepower is that when on sea trials a ship must make its sustained speed using only 80% of the installed horsepower. Most designers like to include an extra 5% for a design margin; this means that a good guideline for SHP_I is:

$$SHP_I = 1.25 \text{ SHP}$$

2.7 <u>POSSIBLE PROPULSION PLANTS</u> Once SHP_I is established, the plant-types to be analyzed can be selected. Again the design requirements, constraints and philosophy can be measured against the plant characteristics to determine the best plants to investigate. Appendix I serves as a propulsion plant shopping list; it shows the functional relationships and gives some of the advantages and disadvantages.

In actual propulsion plant designs, several variations within a single plant-type will be investigated. For instance, a COGAG plant can have many different combinations of GT's, with different initial costs, availabilities, SFC's, maintenance requirements, etc. One of the variations within each plant type is the most efficient (or design) RPM of the prime mover. In as much as most reduction gears are of standard size, or at least limited in range, these prime mover RPM's will not always match the previously calculated propeller RPM. This of course may necessitate another propeller analysis, with an RPM constraint.

By the end of this phase of the design there will be several plant types ready to be sized. Figure 2.4 illustrates the steps that have been described so far.

2.8 <u>PERFORMANCE AND SIZING</u> Each of the plants selected to analyze must be sized and their performance evaluated. In

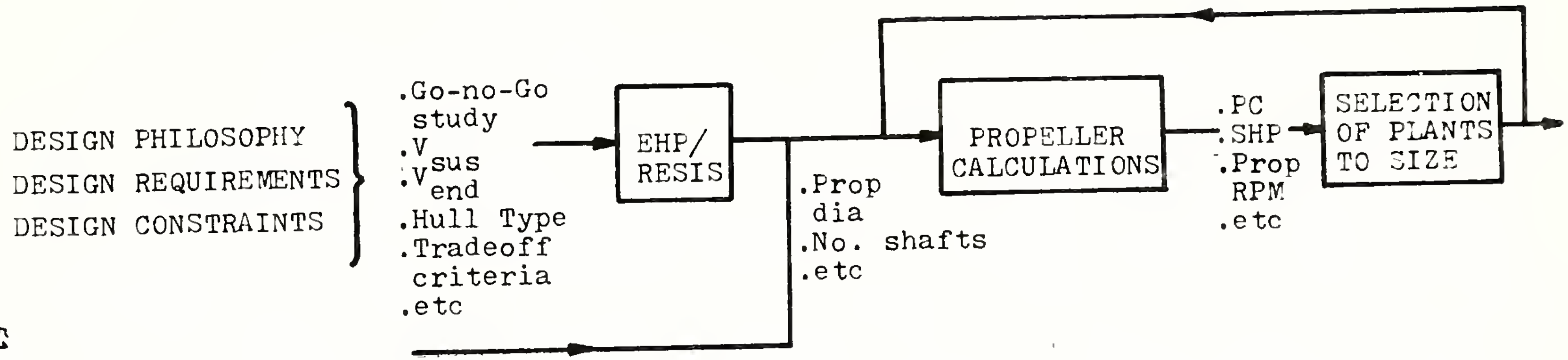

DESIGN PHILOSOPHY
DESIGN REQUIREMENTS
DESIGN CONSTRAINTS
.Go-no-Go study
.V sus
.V end
.Hull Type
.Tradeoff criteria
.etc
EHP/ RESIS
.Prop dia
.No. shafts
.etc
PROPELLER CALCULATIONS
.PC
.SHP
.Prop RPM
.etc
SELECTION OF PLANTS TO SIZE

FIGURE 2.4

this context, sizing pertains to weight and volume and performance usually refers to efficiency.

The weight and volume in the first iteration comes from manufacturer's specifications and emperical data. It normally is restricted to a few weight groups, but since these are high impact areas the relative worth of the results is significant. During the sizing of a specific plant it may become evident that it is too big (based on known constraints and requirements); in this case further study is unwarrented. In fact, at any one point in the design process a component or plant type could be dropped from further consideration based on new information. But the designer must be careful not to eliminate a feasible plant (or a piece of gear) because of poor qualities in only one area. In fact weight and volume should be integrated into an overall ship design feedback system to insure the options for trading off propulsion plant requirements with payload, displacement, volume, etc. are available (and vice versa): See Figure 2.5. Appendix IV contains enough sizing information to gain a very good feel for the weight and volume of the standard plant types.

Performance analyses usually consist of calculating SFC vs Power, reliability/availability and other miscellaneous factors that measure system performance. The SFC data, coupled with projected operating profiles, is used

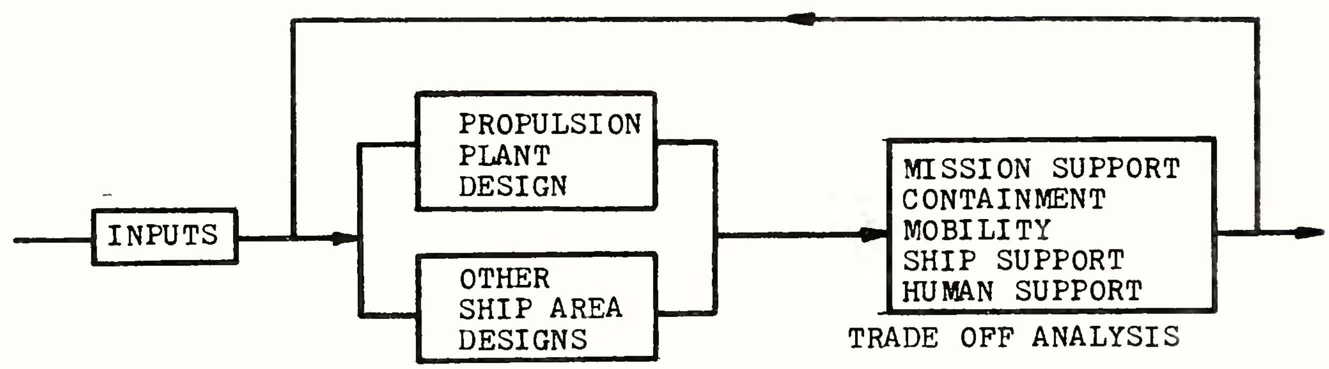


FIGURE 2.5 DESIGN FEEDBACK LOOP

to determine the weight and volume that must be allotted for fuel. The reliability/availability values are dependent on the functional relationships, and redundancies of the various components. Therefore, performance analyses are closely linked to machinery arrangements.

2.8.1 PLANT EFFICIENCY One very important factor in a performance evaluation is plant efficiency. Matching prime mover efficiencies to required SHP, waste-heat recovery systems and auxiliary interfacing are the three prime areas in any efficiency analysis.

As a matter of fact, combined plants are an outgrowth of attempts to optimize SFC at both endurance and maximum sustained speeds. Although Section 1.4 and Appendix I discuss the most popular combined plants, almost anything is feasible. The designer has a real opportunity for innovation in this area.

Overall plant efficiency can also be increased by careful integration of propulsion and auxiliary plants. The best way to demonstrate these options is through a schematic. Figure 2.6 shows two different ways to supply several auxiliary loads in a diesel propulsion plant. Each method has its advantages and disadvantages, excluding effeciency, which have to be considered. How to increase

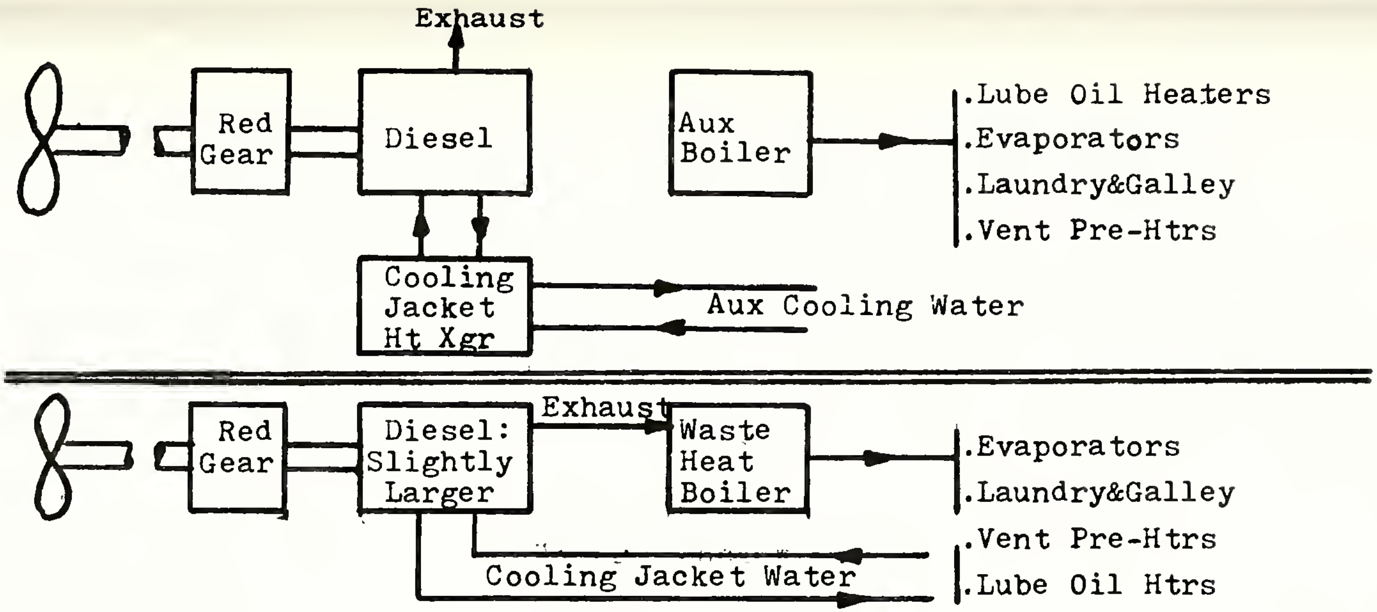


FIGURE 2.6 TWO METHODS OF SUPPLYING AUXILIARY LOADS

efficiency through interfacing will be a consequence (in most instances) of the plant selected, rather than a driving force in the selection process; for this reason it will not be addressed further in this thesis.

2.9 SUPPORTING STUDIES Supporting studies in such areas as manning, technical risk, maintenance, control, noise signature, support, etc. are crucial to the candidate selection process. These studies take place in a series-parallel fashion. They are intradependent (in a limited way) and therefore require continual information exchange. Results from these studies are the basis for assigning Figures of Merit (FOM) to the different candidate plants.

As with other areas of propulsion plant design, supporting studies receive from, and contribute to, the overall ship design feedback system. Some of the supporting studies cannot be realistically assessed until the design progresses to actual component selection; this is especially true in manning and maintenance. But these areas can be estimated well enough to permit a plant selection analysis to be made with confidence. One study that is important, in that it is usually a constraining item, is Reliability and Availiability.

2.9.1 RMA A very important aspect of propulsion design is Reliability, Maintainability and Availability (RMA). As in most other areas, RMA characteristics are the basis for tradeoff studies. For example, the increased availability resulting from added component redundancy might be weighed against its cost in dollars, time, maintenance requirements, weight and volume.

Before discussing RMA in any detail, it is necessary to define the terms most often used. Quite often these terms are used interchangeably or out of context, which is confusing as well as wrong. The following definitions have been taken from reference 4.

> RElIABILITY, OPERATIONAL: Operational reliability is the reliability demonstrated by an equipment under actual field use. It is the probability that a system will give a specified performance for a given period of time, when used in the manner and for the purpose intended.
>
> MAINTAINABILITY: A characteristic of design and installation which is expressed as the probability that an item will be retained in, or restored to, a specific condition within a given time period, when the maintenance is performed in accordance with prescribed procedures and resources.
>
> AVAILABILITY, INHERENT: The probability that a system, or equipment, when used under stated conditions without consideration for any scheduled or preventative maintenance in an ideal support environment, will operate satisfactorily at any given time: It excludes ready time, preventive maintenance downtime, supply downtime and waiting or administrative downtime.

AVAILABILITY, OPERATIONAL: The probability that a system or equipment, when used under stated conditions and in an actual supply environment, will operate satisfactorily at any given time.

Reliability engineering refers to quantifying reliability. The Navy's major input to reliability engineering is the Maintenance Data Collection System (MDCS). This system, along with others, helps determine the inputs to a quantitative analysis of RMA by supplying statistical data used in calculating MTBF, MTTR, MTBM and MDT.

MTBF - mean time between failure.
MTTR - mean time to repair.
MTBM - mean time between maintenance: MTBM = MTBF when PMS downtime is considered zero.
MDT - mean downtime: Includes supply and administrative downtime.

System reliability requirements are determined by the customer prior to design, or negotiated during designer-customer dialogue in the early stages of design. Either way the end result is a set of RMA specifications covering various operating conditions. These specs. will be used by the designer to help determine optimum system and subsystem models. An RMA analysis is presented in Appendix VII.

2.10 <u>CANDIDATE PLANT SELECTION ANALYSIS</u> The difficulty in selecting an optimum propulsion plant, from among the

candidate systems, is one of the most critical steps in the design plan. There are many types of selection analyses to choose from when attempting to define an optimum plant. All of these processes are fundamentally the same, regardless of their individual variances. Basically they consist of quantifying the desired propulsion plant characteristics and comparing each plant to these standards. A qualitative evaluation scheme then scores each candidate plant in those areas of primary concern to the owner/designer.

Probably the simplest and most widely used selection method is referred to as a Figure of Merit (FOM) analysis: Outlined in detail in Appendix VI. The FOM approach consists of quantifying the relative importance of desired system characteristics and then assigning a plant rating in each of these areas:

SAMPLE CHARACTERISTIC	FOM	PLANT RATING'S PLANT 1	PLANT 2	PLANT 3
Weight	1.0	1.0*	.90	.94
Volume	.95	.93	1.0*	.80
Cost	.88	.75	1.0*	.98
Tech.Risk	.87	1.0*	.94	.82
Req.Manning	.80	1.0*	.84	.98

* Note that the 'best' plant (s) in a specific catagory is (are) rated 1.0 and the other's are rated relative to the best.

The system with the highest accumulative score, achieved by multiplying the FOM'S assigned to each characteristic times the respective plant ratings, can be considered an optimum choice from among the feasible candidates.

The chief arguement of this approach is the enormous amount of bias and prejudice that could surface when rating the characteristics and assigning the FOM's. This is a legitimate criticism but then it is also one that can be countered (to a degree) with a few simple guidelines. Below are listed some ways to reduce the bias.

1. Clearly define the design philosophy, requirements and constraints as they apply to the propulsion plant. This should help to eliminate ambiguities in quantifying the importance of system characteristics.

2. Draw from as many different sources as possible when assessing FOM's.

3. Assign FOM's early in the design process, before prejudices are formed.

2.11 <u>AUXILIARY PLANTS</u> The influence of the auxiliary and electrical plants on propulsion plant selection is not a controlling factor, but it can be significant when considering ship service electrical power generation. Quite obviously a plant selection strongly influences the choice of electric power generation. In a steam plant (nuclear or conventional) it is advantageous to use steam turbines in

the electric plant, for several reasons.

1. Economics: Increasing the size of the boiler to accomodate a few thousand extra horsepower is not that costly (compared to installing an entirely new system, such as a GT or diesel). Also the required supporting systems already exist.

2. Manning: Because propulsion and electric power are provided by the same type of machinery there is no need for additional ratings to perform maintenance and watchstanding requirements.

Some of the same arguements that suggest steam plants utilize steam turbines for electric power, apply to gas turbine and diesel plants. For combined plants (GT's and diesels) there are tradeoffs to consider; among them are such things as the lower SFC's of the diesels and weight and maintenance advantages of the GT's.

For a feasibility study, the auxiliary plant, beyond ship's service electric power, need not be considered in propulsion plant design under normal circumstances.

Most synthesis models used in sizing machinery boxes (such as those in references 5 and 6) take into account these close relationships between the propulsion plant and the electric plant. This is reflected in their emperical formulas relating the installed KW to machinery box volume.

For the above reasons the design methodology outlined in this section uses the installed KW in determining volume for different types of propulsion plants. Figure IV-14 gives KW_I estimates for various displacements, and provides the respective machinery box volume relationships.

2.12 CONCLUSION The propulsion plant design methodology presented in this section is an orderly progression from initial inputs to final outputs. It easily lends itself to a spiral design process, requireing only refined data and greater detail for each iteration. The initial inputs and guidelines evolve from the "overall ship" design requirements, including the design philosophy and constraints. The final outputs are a feasible propulsion plant including arrangements, weights, volumes, operating curves and operational reliabilities.

The entire design process can be viewed as one that addresses three basic questions.

1. How much horsepower is required?
2. Which plants deliver the required horsepower and what are their characteristics? (Weight, Volume, etc.)
3. Of those plants that fulfill the requirements, which one is the optimum?

Determining the horsepower requirements is a straight forward calculation based on initial estimates of displacement and hull form paramaters. Model testing and 'series-analysis' are the two methods used to obtain the required power. The power arrived at through these methods is that necessary to propel the bare hull through the water and thus must be ammended several times prior to arriving at the required installed shaft horsepower.

Once the installed shaft horsepower has been established, selecting the optimum propulsion plant begins. There are usually several alternative plants that can fulfill the requirements, each posessing desireable qualities. The main objective then is two fold: Determine those plants that are reasonable candidates and then conduct a tradeoff analysis of the candidate plants to select the optimum one. The tradeoff analysis is basically a weighted comparison of the different plant's characteristics, and therefore requires sizing and performance information on the candidate plants.

This section outlines the procedure for answering the three design questions. This together with the appendices and example design (Section 3) should enable the reader to conduct his propulsion plant design study. Figure 2.7 shows the entire process.

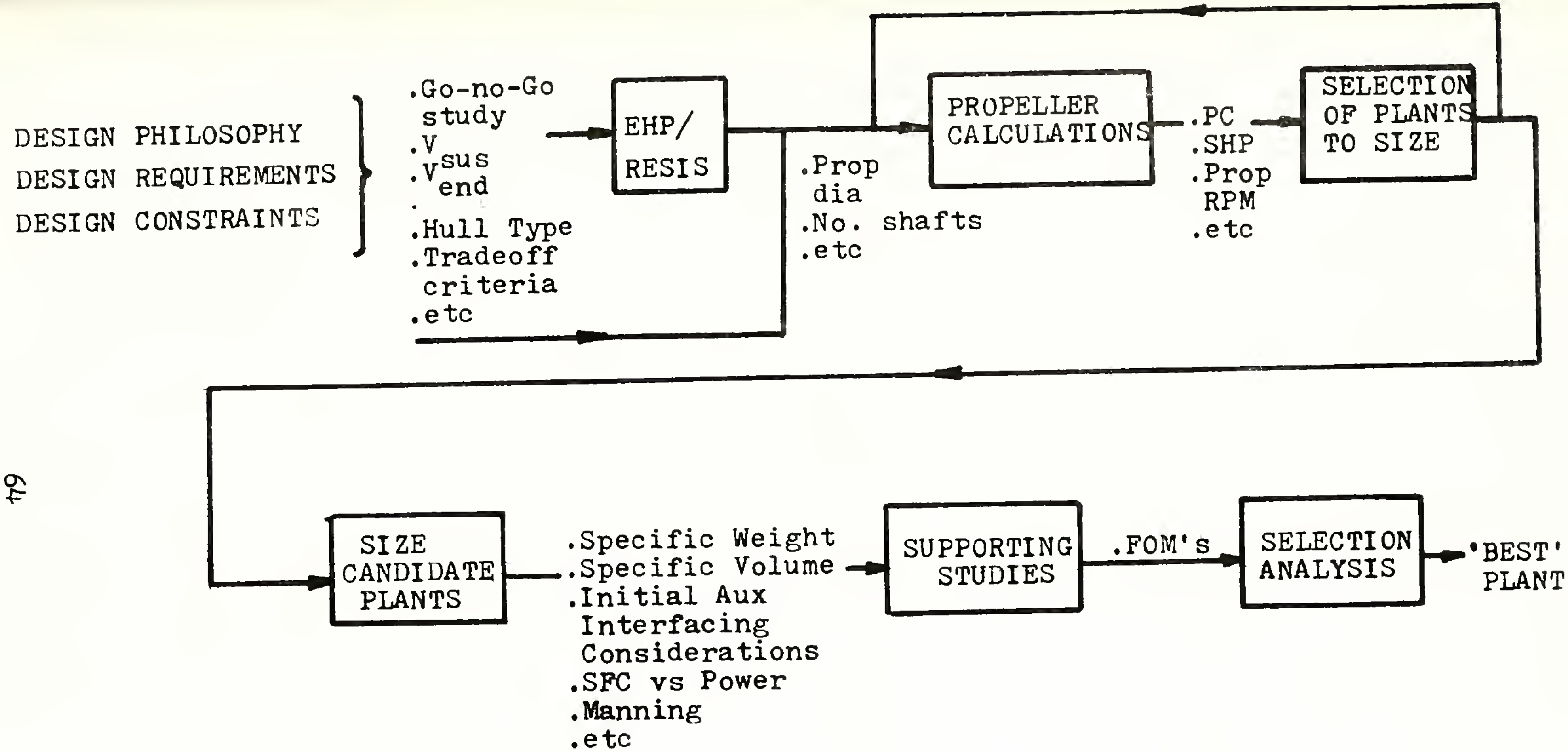


FIGURE 2.7 PROPULSION DESIGN FLOW CHART

3. SAMPLE PLANT SELECTION

3.1 INTRODUCTION This section presents a sample propulsion plant feasibility study using the methodology outlined in Section 2; as well as component data from, and methods shown in, the appendicies.

Beginning with a list of typical inputs, the end result is the selection of a 'best' plant from among those chosen to analyze. Comments on the procedure and reasoning behind the analysis, along with references to formulas and data used, are found throughout the example.

3.2 INPUTS The propulsion plant design inputs are drawn from various sources, as stated in Section 2.4. It is assumed that the hull form has been selected and model tests have been conducted. The results of these tests will provide values for barehull resistance, wake fraction, thrust deduction coefficient and propeller diameter limits.

The propulsion philosophy is simply an interpretation of how the overall ship design philosophy will affect propulsion. This is only the first iteration of a feasibility study and does not have all the inputs that might normally be required to complete the propulsion design. There is, however, enough information to permit the designer to choose, and size, the various plants thought to be competative.

INPUTS TO SAMPLE PROBLEM

INPUT	SOURCE
1. V_{sus} 30kts	design requirement
2. V_{end} 20kts	design requirement
3. Endurance range 6000nm	design requirement
4. System reliability of .95 that at least 50% power is available during a 30 day operation	design constraint
5. Approximate size: Δ .... 7000tons; L .... 525ft; B .... 55ft; T .... 19ft; C_p .... .60; C_x .... .80	design requirement, design constraint, model tests
5. Philosophy: minimize onboard maintenance, time to get underway, technical risk, production time and manning	design requirements, design constraints and design philes-ophy
7. Resistance at V_{sus}.. 277,257lbs	model tests
8. Resistance at V_{end}.. 41,795lbs	model tests
9. 1-w .... .98	Assumed values: see Section 2.6.3
10. 1-t .... .96	
11. Max prop dia. 16ft	Hull form

3.3 PROPELLER SELECTION The propeller selection analysis will assume the sample problem requires two shafts. This is not a bad assumption for a 7000 ton ship required to make 30kts on a 16ft diameter propeller; but more importantly it adds variety to the prime mover selection process. This should add more insight into design reasoning.

The propeller calculations that follow are exactly like those in Appendix II: Using Troost curves from Appendix V. Since the propeller analysis requires the resistance with appendages included, the bare hull resistances must be increased by the ratio EHP_{APP}/EHP_{BH} : Section 2.6.2

RESISTANCES WITH APPENDAGES

$(EHP_{APP}/EHP_{BH})_{30kts} = 1.215$: Figure 2.2

$(EHP_{APP}/EHP_{BH})_{20kts} = 1.3$: Figure 2.2

$(R_{APP})_{30} = 1.215(R_{BH})_{30} = 1.215(277,257)$

$(R_{APP})_{30} = 336,867$ lbs

$(R_{APP})_{20} = 1.3(R_{BH})_{20} = 1.3(41,795)$

$(R_{APP})_{20} = 54,334$ lbs

PROPELLER CALCULATION INPUTS

Resistance at V_{sus} (R) .... 336,867 lbs

Resistance at V_{end} (R) .... 54,334 lbs

Propeller diameter (D) .... 16 ft

V_{sus} .... 30 kts

V_{end} .... 20 kts

No. of shafts (N) 2

1-w .... .98

1-t .... .96

PROPELLER CALCULATIONS

$$\left.\frac{K_t}{J^2}\right|_{30} = \frac{R}{N \cdot D^2 \cdot V^2 \cdot \rho \cdot (1-w)^2 \cdot (1-t)}$$: Equation IIa

$$\left.\frac{K_t}{J^2}\right|_{30} = \frac{336{,}867}{2 \cdot (16)^2 \cdot (30 \times 1.69)^2 \cdot (1.99) \cdot (.98)^2 \cdot (.96)}$$

$$\left.\frac{K_t}{J^2}\right|_{30} = 0.140$$

$$\left.\frac{K_t}{J^2}\right|_{20} = \frac{R}{N \cdot d^2 \cdot V^2 \cdot \rho \cdot (1-w)^2 \cdot (1-t)}$$

$$\left.\frac{K_t}{J^2}\right|_{20} = \frac{54{,}334}{2 \cdot (16)^2 \cdot (20 \times 1.69)^2 \cdot (1.99) \cdot (.98)^2 \cdot (.96)}$$

$$\left.\frac{K_t}{J^2}\right|_{20} = 0.051$$

TABLES OF CONSTANT $\frac{K}{J^2}$ VALUES

30kts K_t	J	20 kts K_t	J
.022	0.4	.008	0.4
.035	0.5	.013	0.5
.050	0.6	.018	0.6
.069	0.7	.025	0.7
.090	0.8	.033	0.8
.113	0.9	.041	0.9
.140	1.0	.051	1.0
.169	1.1	.062	1.1
.202	1.2	.073	1.2
.236	1.3	.086	1.3

The next step in the propeller analysis consists of plotting the above relationships on various propeller curves (see Figure II-1) and constructing a table like Table II-2. Table 3.1 is the table for this problem: values of P/D, J, K_t, and η, were taken from the plots of K_t/J^2 on the propeller curves and n, A_p, T, q_t, P_o-P_v and % cavitation came from the formulas and graph on Figure II-2.

Based on the propeller selection criteria discussed in Section 2.5.3 (high efficiency and minimum cavitation), Table 3.1 suggests propellers B5-75 and B5-90 are the best choices. The B5-75 will be used with prime movers requiring reversible pitch propellers (the expanded area ratio must be less than about .78 to permit blade reversal), and B5-90 will be used with reversible prime movers.

3.4 DETERMINE SHP_I Once the propeller efficiencies are

PROP	V	P/D	J	K_t	η_o	n	A_p	$\frac{T}{A_p}$	q_t	$\frac{T}{A_p q_t}$	$P_o - P_v$	$\frac{P_o - P_v}{q_t}$	%cav
B4-55	30	1.40	1.13	.179	.75	165	82.43	14.76	82.39	.18	21.65	.26	15
B4-55	20	1.50	1.38	.097	.76	90	80.42	3.66	27.01	.14	21.65	.80	0
B4-70	30	1.40	1.12	.176	.74	166	104.55	11.64	83.18	.14	21.65	.26	9.5
B4-70	20	1.50	1.40	.100	.76	89	102.54	2.87	26.58	.11	21.65	.81	0
B4-85	30	1.40	1.12	.170	.73	166	126.67	9.61	83.18	.12	21.65	.26	6
B4-85	20	1.55	1.42	.140	.75	87	122.65	2.40	25.74	.09	21.65	.84	0
B4-100	30	1.40	1.11	.170	.71	168	150.80	8.07	84.79	.10	21.65	.26	1.5
B4-100	20	1.60	1.41	.205	.72	88	140.74	2.09	26.16	.08	21.65	.83	0
B5-60	30	1.50	1.24	.210	.75	150	86.46	14.07	71.05	.20	21.65	.30	15
B5-60	20	1.55	1.45	.260	.76	86	86.46	3.40	25.32	.13	21.65	.86	0
* B5-75	30	1.45	1.14	.200	.74	163	110.58	11.00	80.82	.14	21.65	.27	8.5
* B5-75	20	1.62	1.56	.230	.77	109	104.55	2.82	36.08	.08	21.65	.60	0
* B5-90	30	1.45	1.16	.185	.73	161	132.70	9.17	79.26	.12	21.65	.27	4.5
* B5-90	20	1.60	1.48	.120	.75	84	126.67	2.32	24.50	.09	21.65	.88	0
B5-105	30	1.58	1.25	.220	.71	149	148.78	8.18	70.33	.12	21.65	.31	4
B5-105	20	1.60	1.50	.110	.73	83	148.78	1.98	24.10	.08	21.65	.90	0
B6-65	30	1.53	1.25	.220	.74	149	94.50	12.88	70.33	.18	21.65	.31	11
B6-65	20	1.60	1.49	.120	.70	83	92.49	3.18	24.10	.13	21.65	.90	0
B6-80	30	1.48	1.20	.200	.74	155	116.61	10.43	74.71	.14	21.65	.29	7.5
B6-80	20	1.66	1.55	.140	.72	80	110.58	2.66	22.93	.12	21.65	.94	0
B6-95	30	1.57	1.27	.230	.73	147	134.71	9.03	68.91	.13	21.65	.31	5
B6-95	20	1.68	1.58	.240	.73	79	130.69	2.25	22.55	.10	21.65	.96	0

TABLE 3.1 PROPELLER CALCULATION RESULTS

known, the propulsive coefficient (PC), SHP and SHP_I can be determined.

PROPULSIVE COEFFICIENT

$PC = \eta_o \cdot \eta_H \eta_R$: Section 2.6.3, Table 3.1

$PC(CRP) = (.74)(.98)(1) = .724$

$PC(FP) = (.73)(.98)(1) = .715$

SHAFT HORSEPOWER

$$(SHP) = \frac{(R)(V_{kts})}{(PC)(326)}$$:Appendix II.3, step 1

$$(SHP)_{30} = \frac{(336,867)(30)}{(.724)(326)}$$: CRP

$$(SHP)_{30} = 42,818$$: CRP

$$(SHP)_{30} = \frac{(336,867)(30)}{(.715)(326)}$$: FP

$$(SHP)_{30} = 43,357$$: FP

$$(SHP)_{20} = \frac{(54,334)(20)}{(.77)(326)}$$: CRP

$$(SHP)_{20} = 4,329$$: CRP

$$(SHP)_{20} = \frac{(54,334)(20)}{(.75)(326)}$$: FP

$$(SHP)_{20} = 4,444$$: FP

SHAFT HORSEPOWER INSTALLED

SHP_I = 1.25 SHP : Section 2.6.3

SHP_I = 1.25(42,818) : CRP

SHP_I = 53,522 : CRP

SHP_I = 1.25(43,357) : FP

SHP_I = 54,196 : FP

For this example assume the required installed horsepower is 54,000 HP.

3.5 SELECTION OF PLANTS TO STUDY The selection of plant types thought to be feasible, given the design inputs and SHP_I , is the next step of the design problem. See Section 2.7. For this problem the following plant types were chosen to analyze.

1200 PSI STEAM
NUCLEAR POWER
COGOG
CODOG

The reason behind selecting the above plants is quite simply that none of them is an obvious non-candidate. This approach of studying all those plants that seem feasible minimizes the chances of inadvertantly eliminating a 'best' choice. If there is a poor choice included, the design process should uncover it. Another way to select plants to

study, which does not apply in this instance, is based on meeting some specific requirement(s). For instance, if the inputs heavily favor (or severly penalize) a specific plant characteristic, then a plant can be selected (or eliminated) strictly on its advantages (or disadvantages) in that area alone.

Only Diesel,COSAG and COGAS plants (of those listed in Appendix I) are not considered. Elimination of the Diesel plant is based on weight and noise.The other two plants are not really state-of-the-art and would probably not meet the design philosophy of minimizing production time.

3.6 PLANT SIZING The plants selected to study must be sized. This section shows how to find the high impact weights, volumes and component dimensions.

As pointed out in Section 2.11, the electrical and auxiliary plants do not impact the feasibility of a propulsion plant except in the case of steam. For the NUCLEAR and 1200 PSI plants, it is assumed that electric power is supplied by steam turbines. The NUCLEAR plant group 200 weight (W_{200}) in Figure IV-1 indirectly accounts for this fact, since it is based on displacement. The 1200 PSI steam plant will have to have the ship's service generator's horsepower added to the SHP_I when determining W_{200}.

Determination of endurance fuel weight for the steam plants will be calculated using the specific fuel consumption (SFC) values for steam turbines (Figure V-1), while the combined plants will use SFC's of the appropriate size diesels.

3.6.1 NUCLEAR PLANT The steam turbine weights in Figure IV-4 do not include the reduction gear, so for the first step in sizing this plant is to size it. The bull gear diameter will also be calculated since it is the controlling dimension for reduction gear volume, and is critical in machinery arrangements.

REDUCTION GEAR SIZING

Assume each shaft has a single input turbine connected to a conventional double reduction locked-train reduction gear.

INPUT		SOURCE
Turbine RPM	7800	Typical steam propulsion turbine RPM
K-factors: 1^{st} red	140	Appendix III.1
2^{nd} red	110	Appendix III.1
Propeller speed	161RPM	Table 3.1
SHP_I(per-shaft)	27,000HP	Section 3.4

(1) REDUCTION RATIOS

$$R_2 = \left(\frac{N_{in}}{N_{prop}}\right)^{\frac{1}{2}} + 3 \quad : \text{Appendix III.4}$$

$$R_2 = \left(\frac{7800}{161}\right)^{\frac{1}{2}} + 3 = 9.96$$

$$R_1 = \frac{N_{in}}{N_{out}}$$

$$R_1 = \frac{7800}{(161)(9.96)} = 4.86$$

(2) GEAR WEIGHT

$$W_1 = \frac{(12.95)(SHP_{in})(R_1+1)^3}{(N_{in})(2R_1)(k_1)}$$; Equation IIIe

$$W_1 = \frac{(12.95)(27,00)(5.86)^3}{(7800)(2)(4.86)(140)}$$

$$W_1 = 6.63 \text{ tons}$$

$$W_2 = \frac{(12.95)(SHP_{in})(R_2+1)^3}{(N_{in})(R_2)(K_2)}$$

$$W_2 = \frac{(12.95)(13,500)(10.96)^3}{(7800/4.86)(9,96)(110)}$$

$$W_2 = 130.9 \text{ tons}$$

(3) TOTAL WEIGHT OF TWO REDUCTION GEARS

$$W_{red\ gr} = 2W_2+4W_1 = 2(130.9)+4(6.63)$$

$$W_{red\ gr} = 288.32 \text{ tons}$$

(4) BULL GEAR (2nd reduction gear) DIAMETER

$$d_2^3 = \frac{126{,}050(SHP_{in})(R+1)}{N_{in}(2.25)(R)(K)} \quad ; \text{ equation IIId}$$

$$d_2 = \left[\frac{126{,}050(13{,}500)(10.96)}{(7800/4.86)(2.25)(9.96)(110)}\right]^{1/3}$$

$$d_2 = 16.72 \text{ in}$$

$$D_2 = R_2\, d_2 = (16.72)(9.96) = 166.53$$

$$D_2 = 13.88 \text{ ft}$$

FIND GROUP 2 WEIGHTS FROM APPENDIX IV

COMPONENT	WEIGHT	SOURCE
Reduction gears	288.32	Above calculation
Primary plant	1500.00	Figure IV-1
Turbine	139.29	Figure IV-4
Propellers	26.40	Figure IV-7
Shaft	98.00	Figure IV-6
Bearings	18.66	Figure IV-7
Support sys	156.00	Figure IV-9
Cond&A.E.	52.50	Figure IV-5
TOTAL	2279.17 tons	

DETERMINE PROPULSION PLANT VOLUME

FUNCTIONAL AREA	VOLUME	SOURCE
Machinery box	194,115	Figure IV-14
TOTAL	194,115 ft^3	

MANNING (WATCHSTANDERS)

WATCHSTATION	NO.	SOURCE
EOOW	1	Manning documents
Electric plant operator	2	
Reactor plant operator	2	
Auxiliary electrician	2	
Leading machinest	2	
Upper levelman	2	
Lower levelman	2	
Throttleman	2	
Reactor controls	2	
Reactor support	2	
Radiation technician	1	
TOTAL	20	

NUCLEAR PLANT SUMMARY

Weight	2279.17 tons
Volume	141,279 ft^3
Manning	20

3.6.2 1200 PSI STEAM PLANT Since the graph of 1200 PSI turbine weight in Figure IV-4 includes reduction gears, no reduction weight calculation is required. As stated in Section 3.6 the HP required for the ship's service turbine generators (SSTG's) is needed to determine W_{200} for this plant.

$$HP_{SSTG's} = \left[\frac{1 \text{ HP}}{.745 \text{ KW}}\right]\left[\frac{KW_I}{\text{Cycle Eff.}}\right]$$

Typical steam cycle effeciency is about .33 and KW_I = 2500 KW (Figure IV-14)

$$HP_{SSTG's} = \left[\frac{1}{.745}\right]\left[\frac{2500}{.33}\right]$$

$$HP_{SSTG's} = 10,169$$

The value to use with Figure IV-1 when determing W_{200}, for the 1200 PSI plant, is $SHP = SHP_I + HP_{SSTG}$

$$SHP = 54,000+10,169 = 64,169 \text{ SHP}$$

$$W_{200} = 174 \text{ tons} \qquad \text{;Figure IV-1}$$

SIZE REDUCTION GEAR (BULL GEAR DIAMETER)

Assume the same turbine RPM as in 3.6.1, but that this is a split turbine with two inputs into a double reduction gear; each input provides 13,500 HP. This means that the horsepower of each second reduction pinion is 13,500($\frac{1}{2}$) HP = 7,250 HP.

$$d_2^3 = \frac{126,050(SHP_{in})(R+1)}{N_{in}(2.25)(R)(K)}$$

$$d_2 = \left[\frac{(126,050)(7,250)(10.96)}{(7800/4.86)(2.25)(9.96)(110)}\right]^{\frac{1}{3}}$$

$$d_2 = 13.59 \text{ in}$$

$$D_2 = R_2\, d_2 = (13.59)(9.96) = 135.39 \text{ in}$$

$$D_2 = 11.28 \text{ ft}$$

Comparing this bullgear to the one for the NUCLEAR plant shows the advantage of two power inputs to a double reduction gear.

FIND GROUP 2 WEIGHTS FROM APPENDIX IV

COMPONENT	WEIGHT	SOURCE
Turbine&red gr	233.84	Figure IV-4
Propellers	26.40	Figure IV-7
Shaft	98.00	Figure IV-6
Bearings	18.66	Figure IV-7
Support sys	156.00	Figure IV-9
Cond&A.E.	52.50	Figure IV-5
Boiler (W_{200})	174.00	Figure IV-1
Uptakes	14.00	Figure IV-8
Total	773.4 tons	

CALCULATE ENDURANCE FUEL WEIGHT

The specific fuel consumption of the propulsion turbines is interpreted, from Figure V-1 , to be .438lbs/hp-hr and .46 for the SSTG's. Also cruise KW is assumed to be .33 KW_I.

$$W_f \text{ (endurance fuel weight)} = (SHP)_{20}(SFC)\left[\frac{\text{Range}}{V_{end}}\right]\left[\frac{1 \text{ ton}}{2240 \text{ lbs}}\right] + (.33)(HP_{SSTG's})(.46)\left[\frac{\text{Range}}{V_{end}}\right]\left[\frac{1 \text{ ton}}{2240 \text{ lbs}}\right]$$

$$W_f = (4{,}444)(.438)\left[\frac{6000}{20}\right]\left[\frac{1}{2240}\right] + (.33)(10{,}169)(.46)\left[\frac{6000}{20}\right]\left[\frac{1}{2240}\right]$$

$$W_f = 467.43 \text{ tons}$$

MANNING (WATCHSTANDERS)

WATCHSTATIONS	NO.	SOURCE
EOOW	1	Manning documents
ACC console operator	2	Fireroom
Burnerman	2	Fireroom
Messenger	2	Fireroom
Upperlevelman	2	Fireroom

WATCHSTATIONS	NO.	SOURCE
Throttleman	2	Engineroom
Upperlevelman	2	
Lowerlevelman	2	
Messenger	2	
Electrician	1	
TOTAL	18	

DETERMINE PROPULSION PLANT VOLUME

FUNCTIONAL AREA	VOLUME	SOURCE
Machinery box	129,676	Figure IV-14
Uptakes	9,800	Figure IV-10
TOTAL	139,476 ft^3	

1200 PSI PLANT SUMMARY

Weight (including W_f)	1240.83 tons
Volume	139,476 ft^3
Manning	18

3.6.3 COGOG PLANT Because of their poor off-speed SFC's (Section 1.4), gas turbines of different ratings are often combined (Section 1.3.1) to enhance their overall performance. For this sample problem the combination chosen was an LM2500 and a Lycoming TF35. Their respective horsepower ratings of 20,000 and 2,500 match up well with the required sustained and endurance horsepower. The turbine characteristics are taken from Table V-1

Each LM2500 and TF35 combination will drive a controllable reversible pitch propeller through a double reduction-locked train reduction gear. The reduction gear will be sized to accept the maximum rating of the LM2500: 27,000 HP.

REDUCTION GEAR SIZING

INPUT		SOURCE
Turbine RPM	3600	Table V-1
K-factors: 1^{st} red	140	Appendix III.1
2^{nd} red	110	Appendix III.1
Propeller speed	163RPM	Table 3.1

(1) REDUCTION RATIOS

$$R_2 = \sqrt{\frac{N_{in}}{N_{prop}}} + 3$$: Appendix III.4

$$R_2 = \sqrt{\frac{3600}{163}} + 3 = 7.7$$

$$R_1 = \frac{N_{in}}{N_{out}}$$

$$R_1 = \frac{3600}{(163)(7.7)} \doteq 2.87$$

(2) GEAR WEIGHT

$$W_1 = \frac{(12.95)(SHP_{in})(R_1+1)^3}{(N_{in})(2R_1)(K_1)}$$: Equation IIIe

$$W_1 = \frac{(12.95)(27,000)(3.87)^3}{(3600)(5.74)(140)}$$

$$W_1 = 7.01 \text{ tons}$$

$$W_2 = \frac{(12.95)(SHP_{in})(R_2+1)^3}{(N_{in})(R_2)(K_2)}$$

$$W_2 = \frac{(12.95)(13,500)(8.7)^3}{(3600/2.87)(7.7)(110)}$$

$$W_2 = 108.36 \text{ tons}$$

(3) TOTAL WEIGHT OF TWO REDUCTION GEARS

$$W_{red\ gr} = 2W_2+4W_1 = 2(108.36)+4(7.01)$$

$$W_{red\ gr} = 244.76$$

(4) BULL GEAR DIAMETER

$$d_2^3 = \frac{126,050(SHP_{IN})(R_2+1)}{N_{in}(2.25)(R_2)(K)} \quad ; \text{ Equation IIId}$$

$$d_2 = \left[\frac{(126,050)(7,250)(8.7)}{(3600/2.87)(2.25)(7.7)(110)}\right]^{\frac{1}{3}}$$

$$d_2 = 14.89 \text{ in}$$

$$D_2 = d_2\ R_2 = (14.89)(7.7) = 114.63 \text{ in}$$

$$D_2 = 9.55 \text{ ft}$$

FIND GROUP 2 WEIGHTS FROM APPENDIX IV

COMPONENT	WEIGHT	SOURCE
Reduction gears	244.76	Above calculation
Turbines	11.33	Table V-1
Propellers	36.68	Figure IV-7
Shaft	196.00	Figure IV-6
Bearings	34.90	Figure IV-7
Uptakes	20.80	Figure IV-8
Lube oil	13.00	Figure IV-9
TOTAL	557.47 tons	

DETERMINE PROPULSION PLANT VOLUME

FUNCTIONAL AREA	VOLUME	SOURCE
Machinery box	122,007	Figure IV-14
Uptakes	24,000	Figure IV-10
TOTAL	146,007 ft^3	

CALCULATE ENDURANCE FUEL WEIGHT

The SFC of the TF35 operating at SHP_{end} (87% normal rating) is .59 lbs/hp-hr: See Figure V-2. The electric plant requirement is assumed to be supplied by high speed diesels with SFC's of .375 (a good assumption based on Figure V-2).

$$W_f = (SHP)_{20}(SFC)\left[\frac{Range}{V_{end}}\right]\left[\frac{1\ ton}{2240\ lbs}\right]$$

$$+ (.33)(HP_{SSTG's})(.375)\left[\frac{Range}{V_{end}}\right]\left[\frac{1\ ton}{2240\ lbs}\right]$$

$$W_f = (4,329)(.59)\left[\frac{6000}{20}\right]\left[\frac{1}{2240}\right] + (.33)(10,169)(.375)\left[\frac{6000}{20}\right]\left[\frac{1}{2240}\right]$$

$$W_f = 510.61 \text{ tons}$$

MANNING (WATCHSTANDERS)

WATCHSTATIONS	NO.	SOURCE
EOOW	1	Manning documents
Propulsion console	1	
Auxiliary console	1	
Electric console	1	
TOTAL	4	

COGOG PLANT SUMMARY

Weight (including W_f)	1068.08 tons
Volume	146,007 ft^3
Manning	4

3.6.4 CODOG PLANT The CODOG plant differs from the COGOG plant only in the choice of prime movers for endurance, and lower, speeds. This plant uses a more fuel efficient diesel for low power requirements. If LCC were included in the selection analysis this would be an even more desireable tradeoff.

For this sample problem the CODOG plant consists of one LM2500 and one Fairbanks Morse 9 cylinder diesel(2700BHP) coupled to each shaft. Since high speed operations will be provided by the LM2500, this plant has the same reduction gear as the COGOG plant.

FIND GROUP 2 WEIGHTS FROM APPENDIX IV

COMPONENT	WEIGHT	SOURCE
Reduction gears	244.76	Section 3.6.3
Turbines	10.36	Table V-1
Diesels	62.68	Figure IV-3
Propellers	36.68	Figure IV-7
Shaft	196.00	Figure IV-6
Bearings	34.90	Figure IV-7
Uptakes	28.50	Figure IV-8
Lube oil	21.00	Figure IV-9
TOTAL	634.88 tons	

CALCULATE ENDURANCE FUEL WEIGHT

The off power SFC of the diesel is taken from Figure V-6. The value .38 lbs/hp-hr represents an 80% load ((4,329/5400). The electric power source is the same as it was for the COGOG plant.

$$W_f = (SHP)_{20}(SFC)\left[\frac{Range}{V_{end}}\right]\left[\frac{1\ ton}{2240\ lbs}\right]$$

$$+ (.33)(HP_{sstg's})(.375)\left[\frac{Range}{V_{end}}\right]\left[\frac{1\ ton}{2240\ lbs}\right]$$

$$W_f = (4,329)(.38)\left[\frac{6000}{20}\right]\left[\frac{1}{2240}\right] + (.33)(10,169)(.375)\left[\frac{6000}{20}\right]\left[\frac{1}{2240}\right]$$

$$W_f = 388.85 \text{ tons}$$

This represents a savings of 121.76 tons over the COGOG fuel weight.

MANNING (WATCHSTANDERS)

WATCHSTATIONS	NO.	SOURCE
EOOW	1	Manning documents
Propulsion console	1	
Auxiliary console	1	
Electric console	1	
TOTAL	4	

DETERMINE PROPULSION PLANT VOLUME

FUNCTIONAL AREA	VOLUME	SOURCE
Machinery box	122,007	Figure IV-14
Uptakes	24,000	Figure IV-10
TOTAL	146.007 ft^3	

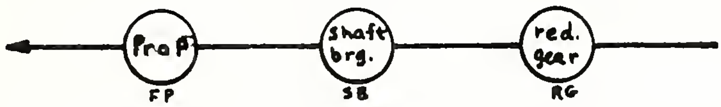
CODOG PLANT SUMMARY

Weight (including W_f)	1023.75 tons
Volume	146,007 ft^3
Manning	4

3.7 RELIABILITY The reliability calculations are like the sample in Appendix VII, using values of MTTR and MTBF from Tables VII-1 and VII-2. They are arranged in subsystems with the combined values calculated last. The design input for reliability is interpreted as meaning : The reliability of at least one shaft available (at maximum SHP) during a 30 day operation is 0.95

CALCULATE THE RELIABILITY OF TRANSMISSION AND PROPULSOR (R_1)

FIXED PITCH PROPELLER (FP)



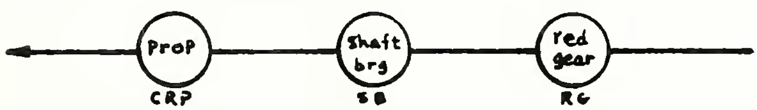
$R_{FP} = 1-\lambda\, t = 1- \frac{720}{200{,}000} = .9964$; Equation VIIa

$R_{SB} = 1-\lambda\, t = 1- \frac{720}{200{,}000} = .9964$

$R_{RG} = 1-\lambda\, t = 1- \frac{720}{200{,}000} = .9964$

$R_{1\,FP} = R_{FP}\, R_{SB}\, R_{RG} = .9892$; Equation VIIb

CONTROLLABLE REVERSIBLE PITCH PROPELLER (CRP)



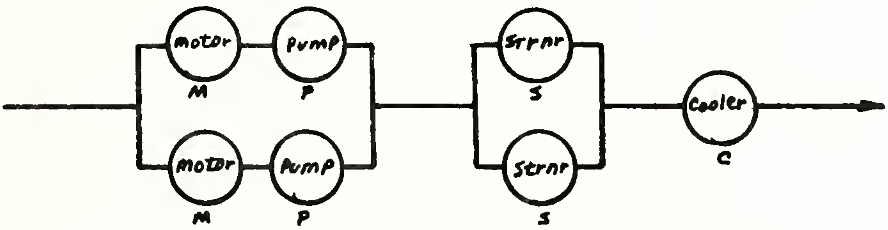
R_{SB} and R_{RG} are the same values as above

$R_{CRP} = \frac{\mu + \lambda e^{-(\mu+\lambda)t}}{\mu + \lambda}$; Equation VIIc

$R_{CRP} = \frac{.067+4\text{x}10^{-5}\exp(-.06704 \cdot 720)}{.06704} = .999$

$R_{1CRP} = R_{CRP}\, R_{SB}\, R_{RG} = .9927$

CALCULATE THE RELIABILITY OF REDUCTION GEAR LUBE OIL SYSTEM (R_2)



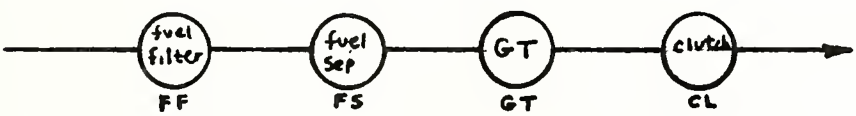
R_M	= .999	Table VII-1
R_P	= .999	Table VII-1
R_S	= 1.0	Table VII-1
R_C	= 1.0	Table VII-1

$$R_2 = (R_M R_P + R_m R_P - R_m^2 R_p^2)(2R_S - R_S^2)(R_C) \quad ; \text{ Section VII-2}$$

$$R_2 = 1.0$$

CALCULATE THE RELIABILITY OF THE PRIME MOVERS (R_3)

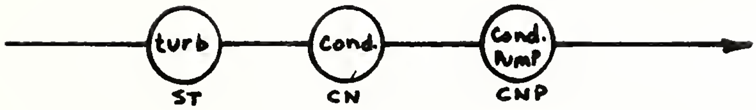
GAS TURBINE (GT)



R_{FF}	= 1.0	Table VII-1
R_{FS}	= 1.0	Table VII-1
R_{GT}	= .994	Table VII-1
R_{CL}	= .986	Table VII-1

$$R_{3GT} = R_{FF} R_{FS} R_{GT} R_{CL} = .98$$

STEAM TURBINE (ST)



$R_{ST} = 1.0$ Table VII-1

$R_{CN} = 1.0$ Table VII-1

$R_{CNP} = 1.0$ Table VII-1

$$R_{3ST} = R_{ST}\, R_{CN}\, R_{CNP} = 1.0$$

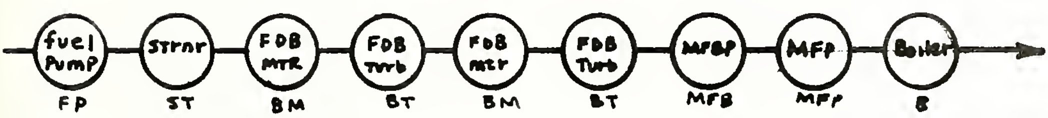
CALCULATE THE RELIABILITY OF ENERGY CONVERTERS FOR THE STEAM PROPULSION PLANTS (R_4)

NUCLEAR (N)

There is no information available on the nuclear primary plant, so it is assumed to be 0.999

$$R_{4N} = .999$$

CONVENTIONAL BOILER SYSTEM (C)



$R_{FP} = .9956$ Table VII-1

$R_{ST} = 1.0$ Table VII-1

$R_{BM} = .9986$ Table VII-1

$R_{BT} = .9976$ Table VII-1

$R_{MFB} = .9972$ Table VII-1

$R_{MFP} = .9944$ Table VII-1

$R_B = .9988$ Table VII-1

$$R_{4C} = R_{FP}\, R_{ST}\, R_{BM}\, R_{BT}\, R_{MFB}\, R_{MFP}\, R_B = .982$$

CALCULATE PLANT RELIABILITIES (R_5)

The plant reliabilities are simply the product of their respective subsystem reliabilities.

NUCLEAR

$$R_N = R_{4N} R_{3ST} R_2 R_{1FP}$$

$$R_N = (.999)(1.0)(1.0)(.9892)$$

$$R_N = .998$$

1200 PSI STEAM

$$R_{1200} = R_{4C} R_{3ST} R_2 R_{1FP}$$

$$R_{1200} = (.982)(1.0)(1.0)(.9892)$$

$$R_{1200} = .971$$

COGOG/CODOG

Since both of these plants have the same machinery configuration at maximum power, their reliabilities will be the same

$$R_{CG/CD} = R_{3GT} R_2 R_{1CRP}$$

$$R_{CG/CD} = (.98)(1.0)(.9927)$$

$$R_{CG/CD} = .973$$

The result of the reliability analysis demonstrates that each plant meets the reliability requirement for this sample problem.

3.8 <u>FIGURE OF MERIT ANALYSIS</u> The key plant characteristics and their FOM's should represent the areas of primary importance to the customer/user. Relative ranking and value assignments for this sample problem are described below: See Appendix VI.

3.8.1 <u>FOM ASSIGNMENTS</u>

VOLUME: FOM = 1.0

Plots of ship density vs year commissioned (7) show the trend towards volume limited ships. The reasons behind this vary from changes in habitability and maintenance philosophies to technical advances in all design areas. In any case it is considered the most important design characteristic in this problem.

WEIGHT: FOM = 0.95

In any ship design with a limit on full load displacement, weight is a key factor. Because weight saved in propulsion is weight available for payload. Also, ship acquisition cost is proportional to full load displacement, for similar ships. For these reasons weight is considered next to volume in importance.

OPERABILITY/MAINTAINABILITY : FOM = 0.90

There has been an increased emphasis placed on this area in U.S. ship design, and therefore, it must be considered an important plant characteristic. Operability refers to ease of operation, complexity of automation required and the number of men needed to operate the plant. Maintainability is a measure of the ease with which maintenance can be performed.

MANNING (WATCH STANDERS): FOM = 0.85

Reduction of manning requirements is important in today's Navy because of its impact on cost and habitability. But since this study looks at only watchstanders, the FOM assignment is not as high as would be if total manning were considered.

RESPONSE TIME: FOM = 0.80

Response time is a measure of the ships' ability to get underway quickly and respond to power changes. This characteristic becomes increasingly important in an age of sophisticated weapons.

RADIATED NOISE: FOM = 0.70

Because ASW is normally a mission area for all cruisers and destroyers, some consideration must be given to the noise generated by the propulsion plant.

Although there are many other plant characteristics that are important (cost, damage, vulnerability, ILS, etc.), the ones listed will serve as a good illustrative guide for a Figure of Merit analysis.

3.8.2 <u>RELATIVE PLANT RATINGS</u> Each plant must be rated (relative to each other) in those areas for which an FOM value has been assigned: Appendix VI.2. For weight, volume and manning, the rating assigned to the 'best' plant (in these areas) is 10; and the other plants ratings represent their relative comparisons to that 'best' plant: Appendix VI.2.

For the remaining areas (operability/maintainability, response time, radiated noise), each plant is rated using subjective reasoning as shown below:

VOLUME

PLANT	RATING
1200 Psi	10
Nuclear	9.9
COGOG	9.6
CODOG	9.6

WEIGHT

PLANT	RATING
CODOG	10
COGOG	9.6
1200 Psi	8.3
Nuclear	4.5

MANNING

PLANT	RATING
CODOG	10
COGOG	10
1200 Psi	2.2
Nuclear	2.0

OPERABILITY/MAINTAINABILITY

All of the plants in this problem are capable of automation. The Nuclear plant has so many paramaters to monitor that its controls are both complicated and numerous. On the other extreme the COGOG and CODOG plants have fewer (and less remote) paramaters to monitor. The nuclear and combined plants are also at opposite ends of the spectrum with respect to the number of support systems. The chief difference between the COGOG and CODOG plants is the increased maintenance requirements of the diesel. In the complexity of the automation and number of support systems, the 1200 Psi plant is closer to the nuclear plant than the COGOG or CODOG. For all these reasons the following ratings have been assigned:

PLANT	RATING
COGOG	10
CODOG	9
1200 Psi	7
Nuclear	6.5

RESPONSE TIME

Gas turbines and diesels coupled to CRP propellers provide the combined plants with virtual instantaneous speed changes and start-ups. Although the nuclear plant can respond to speed changes as fast as the throttleman, it does take an hour or two to start-up, even from hot conditions. Compared to the other plants, the 1200 Psi plant is slower getting underway (at least two hours from cold start up) and responding to speed changes. As a result of the above reasons, the following ratings were assigned:

PLANT	RATING
COGOG	10
CODOG	10
Nuclear	8
1200 Psi	7

RADIATED NOISE

Of the plants chosen to study the CODOG plant is most likely to have the worst self-generated noise level. This is because of the inherent noise level of the diesel. The other plants should be relatively quiet, with the nuclear plant being the best. The assigned ratings are:

PLANT	RATING
Nuclear	10
1200 Psi	9
COGOG	8
CODOG	7

3.8.3 FOM TABLE Once the FOM's and plant ratings have been determined an analysis table such as Table VI-1 can be constructed. The sum of the products of FOM and ships ratings, should show which plant is the 'best'.choice for this sample problem. See Table 3.1.

3.9 CONCLUSION Table 3.1 shows that the COGOG plant is the one best suited for this sample problem: From among the four plants studied. Its slight advantage in operability/maintainability and radiated noise, off-set the weight savings that the more fuel effecient diesel afforded to the CODOG plant. Each of the steam plants fared low because of weight and manning. Had this been a ship with a projected full-load displacement of 3500 tons this weight margin would have been even more of a factor than it was.

This sample problem clearly outlines the initial steps in propulsion plant selection, and shows the importance of weighting (assigning FOM's) selection criteria. For instance if life cycle costs were given a high FOM, the low fuel requirement of the CODOG plant could easily have made it the 'best' choice. The sample also demonstrates that the appendicies of this thesis contain enough information to permit the first iteration of a propulsion plant feasibility study to be made, without consulting further sources.

CHARACTERISTIC	FOM	NUCLEAR		1200 PSI		COGOG		CODOG	
		RATING	PROD	RATING	PROD	RATING	PROD	RATING	PROD
VOLUME	1.00	9.9	9.9	10.0	10.0	9.6	9.6	9.6	9.6
WEIGHT	0.95	4.5	4.28	8.3	7.89	9.6	9.12	10.0	9.5
OP/MAIN	0.90	6.5	5.85	7.0	6.3	10.0	9.0	9.0	8.1
MANNING	0.85	2.0	1.7	2.2	1.87	10.0	8.5	10.0	8.5
RESPONSE	0.80	8.0	6.4	7.0	5.6	10.0	8.0	10.0	8.0
NOISE	0.70	10.0	7.0	9.0	6.3	8.0	5.6	7.0	4.9
TOTAL			35.13		37.96		49.82		48.60
OVERALL RATING			4		3		1		2

TABLE 3.1 PLANT SELECTION ANALYSIS TABLE

4. SUMMARY AND RECOMMENDATIONS

4.1 <u>SUMMARY</u> For almost any proposed ship design there exists more than one feasible propulsion plant. Which of these best meets the design objectives depends on the impact they make in several key areas: Such as volume, weight, manning, etc. The design approach outlined in this thesis consists of determining the required shaft horsepower and the different ways to provide it. Once this is accomplished, those plants chosen to investigate are sized and compared. Then their relative worth in the key areas mentioned above are assessed to determine which one is best suited to the initial design requirements.

Transforming design inputs into feasible plants requires an understanding of propeller design, gear design, reliability calculations and selection analysis.along with sizing information and specific machinery data. All of this information is found in the appendices. There are explicit examples of how propellers are selected, reduction gears sized and system/subsystem reliabilities calculated. All significant Group 2 weights, and their associated volumes, are presented along with information on gas turbines, steam turbines and diesels. Finally there is a sample method on propulsion plant selection analysis.

This thesis presents a methodology and provides supporting data which should enable a basic feasibility study to be made (for state-of-the art propulsion plants) without consulting other references.

4.2 <u>RECOMMENDATIONS</u> The next logical step in propulsion design, which this thesis does not address, is an increased level of detail, preliminary investigation into auxiliary operation and propulsion plant concepts for advanced marine vehicles.

The increased level of detail in the propulsion plant will permit more refined estimates of volume, weight, manning, technical feasibility maintenance requirements, etc. Such things as plant control, degree of automation and maintenance philosophy are the type of factors that will govern this process.

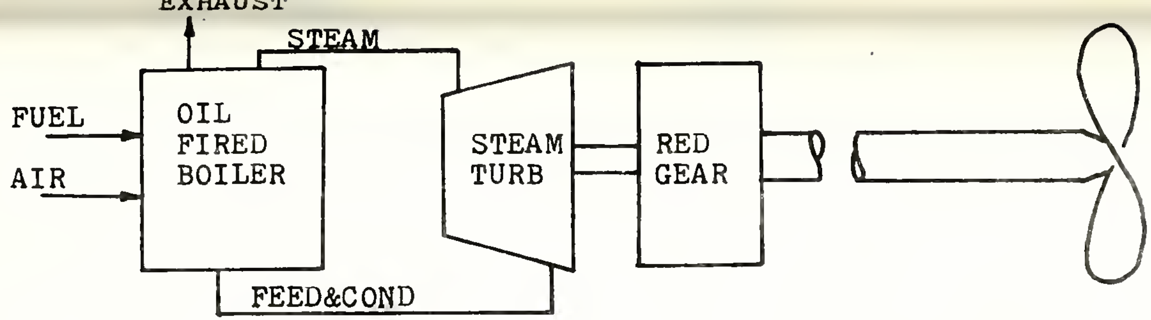
If propulsion plant - auxiliary plant interfacing is used to enhance plant efficiency then this too must be analyzed to refine the propulsion plant paramaters.

Accomplishing the above will require the formulation of a sound methodology and the necessary supporting information. The output of such a scheme is a necessary step in the iteration of a propulsion plant.

APPENDIX I: PROPULSION PLANT SHOPPING GUIDE

I.1 <u>INTRODUCTION</u> APPENDIX I shows the functional relationships and lists the advantages and disadvantages of the standard cruiser/destroyer propulsion plants. It is intended to provide a brief introduction/guide to the basic characteristics of various plants. It can be useful in making the 'go-no-go' decisions referred to in Section 2. or simply as a reminder of the impacts associated with a particular plant selection. The advantages and disadvantages are of a general nature and are not intended to be all inclusive. In fact circumstances may negate the affects of any one characteristic.

I.2 <u>PROPULSION PLANTS LISTED</u> The following plants are found in this appendix.



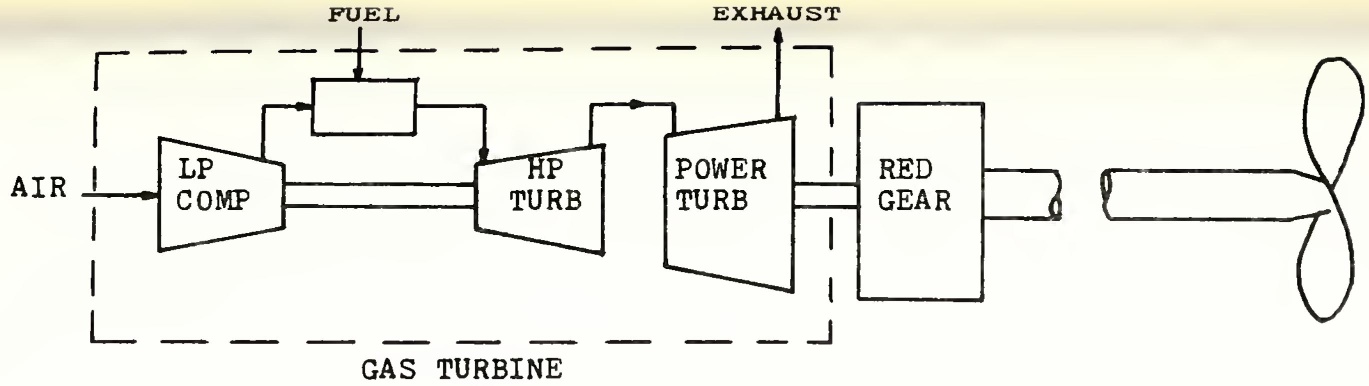
ADVANTAGES

1 Extensive service experience
   a Reliability
   b Parts availability
2 Cycle flexibility
3 Ability to burn low grade fuels
4 Reversible prime mover
5 Easy low speed operations
6 Quiet: low vibration levels
7 Low maintenance requirements
8 Most PMS can be done during operation

DISADVANTAGES

1 Low power density (SHP/VOL)
2 High manning requirements
3 Slow start-ups and reaction times
4 Large number of supporting sub-systems

FIGURE I.1 OIL FIRED STEAM PLANT



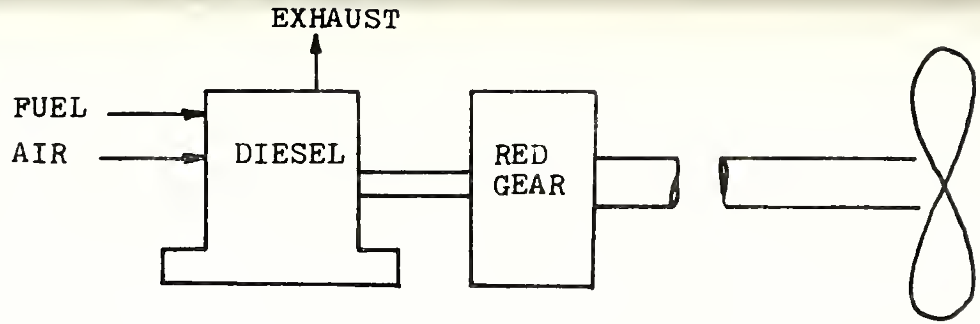
ADVANTAGES

1 High power to weight ratio
2 Compact design
    a Easy installation
    b Ease of removal as a unit
3 Quick response
4 Fast start-up
5 Ease of automation (low manning)
6 Flexible
    a easily adaptable to combined cycle use

DISADVANTAGES

1 Poor off-design SFC
2 Unidirectional
    a Need CRP or reversing gears
3 Sensitive to varying atmospheric conditions and fuel quality
4 High airborne noise signature
    a Requires sound isolation
5 Low power density (SHP/VOL)

FIGURE I.2 GAS TURBINE PLANT



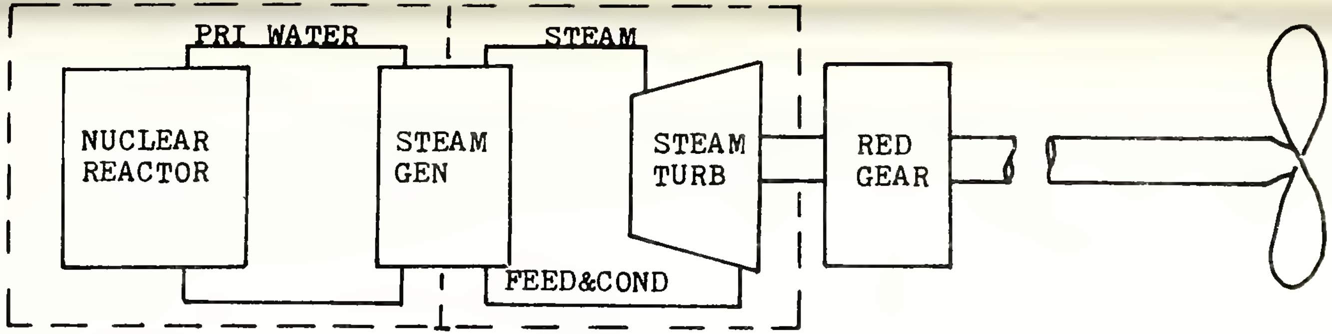
ADVANTAGES

1 Low SFC
2 Ability to burn low grade fuels
3 Low manning levels
4 Easily automated
5 Quick response
6 Fast start-ups

DISADVANTAGES

1 Low power to weight ratio
2 Poor slow speed operations
3 Unidirectional
   a Requires CRP, clutches or stop and restart
4 High noise signature
   a Degrades sonar operations
5 High maintenance requirements
6 High lube oil consumption

FIGURE I.3 DIESEL PLANT



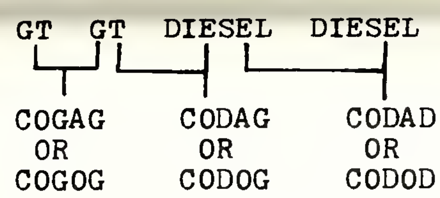
ADVANTAGES

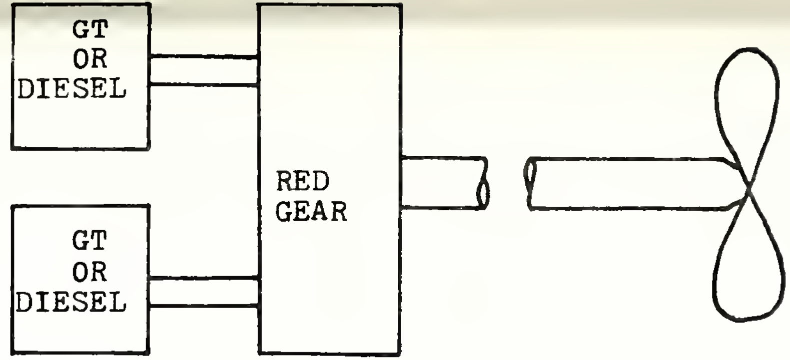
1 Unlimited range and endurance
2 Quick response
3 Low LCC due to long core life
4 Quiet operation
5 No topside impact

DISADVANTAGES

1 Potential health hazard
2 Low power to weight ratio
3 High initial cost
4 Personnel problems
   a Hard to recruit
   b Training expensive

FIGURE I.4 NUCLEAR PLANT





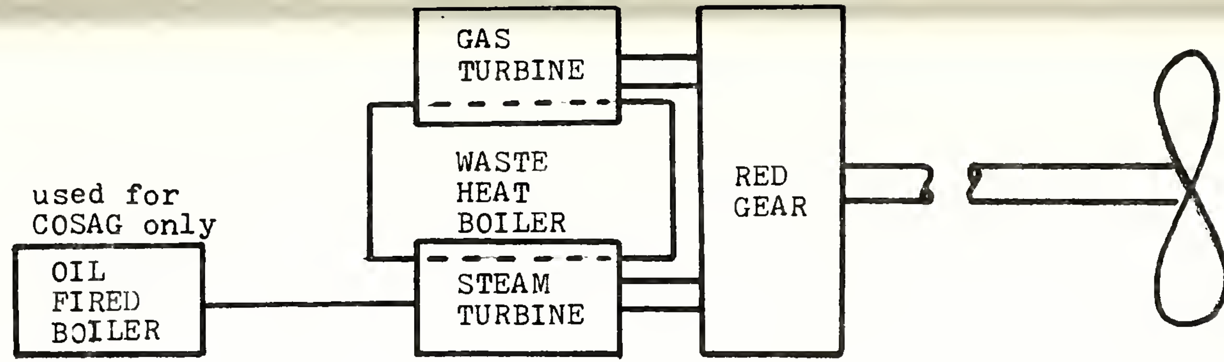
ADVANTAGES

1 Matches required power to power available
2 More efficient use of prime movers
3 Improves SFC at low power requirements
4 Can meet power requirements unattainable by single prime mover

DISADVANTAGES

1 Requires clutching
   a Expensive
   b Technically complicated
2 Automation more difficult
3 Two crew skills required
   a CODAG and CODOG
4 Many of the individual disadvantages are retained: although their impact may be reduced

FIGURE I.5 COMBINED GAS TURBINE AND/OR DIESEL PLANT



.COGAS - Gas turbine power base plus steam turbine power boost from waste heat recovery

.COSAG - Steam turbine power from oil fired boiler plus gas turbine boost

ADVANTAGE

1. Reduced fuel consumption through increased cycle efficiency

DISADVANTAGES

1. Two crew skills required
2. Complex controls
3. Difficult to arrange

FIGURE I.6 COMBINED GAS TURBINE AND STEAM PLANT

APPENDIX II - PROPELLER SELECTION

II.1 LIST OF SYMBOLS

EAR .... Expanded Area Ratio

$J = V_a/n \cdot D$..................Advance Ratio

$K_t = T/\rho \cdot n^2 \cdot D^4$................Thrust Coeffecient

$K_q = Q/\rho \cdot n^2 \cdot D^5$................Torque Coeffecient

$\eta_o = (J/2\pi)(K_t/K_q)$..........Open water propeller effeciency

V ....Ship's Speed

V_a...........................Speed Of Advance

$w = (V-V_a)/V$.................Wake Fraction

T ....Thrust provided by screw

R_t ....Thrust avaible to overcome ship's resistance

$t = 1 - (R_t/T)$..............Thrust Deduction Factor

D ....Propeller Diameter

n ....Propeller rpm

$\eta_h = (1 - t)/(1 - w)$ ....Hull Effeciency

P ....Propeller Pitch

$\eta_r = \dfrac{\text{open water torque}}{\text{actual torque}}$ ......Relative Rotative Effeciency

$PC = \eta_o \cdot \eta_h \cdot \eta_r$ ....Propulsive Coeffecient

$\rho = 1.99$ lb-sec^2/ft^4..... Density of Salt Water

$\tau_c = T/(A_p\ q_t)$ .... Propeller Thrust Loading

$\sigma = \dfrac{p_0 - p_v}{q_t}$ .... Local Cavitation Number

A_p ..... Projected Area of All Blades, Outside Hub

q_t .... Dynamic Pressure

II.2 PROPELLER CALCULATION INPUTS

1. Estimated ship resistance (or EHP_{app}) at various speeds.
   a. These values are determined through model testing or Series Analysis.

2. Maximum allowed propeller diameter (or rpm).
   a. For Navy cruisers and destroyers, the following estimates are good (9).

   $D = 2.60\ H^{0.629}$ one shaft

   $D = 4.28\ H^{0.428}$ two shafts

3. Number of shafts

4. Estimated values for w, t and η_r (see Section 2.6.3)

5. Required speeds (V_{end} and V_{sus})

II.3 EXAMPLE PROPELLER CALCULATION The following example assumes there are to be two shafts and that the propeller diameter is fixed.

INPUTS

Resistance (R)	181,597 lbs (30kts)
	37,064 lbs (18kts)
Propeller diameter (D)	10.5 ft
V_{sus}	30 kts
V_{end}	18 kts
No. of shafts (N)	2
1-w	0.94
1-t	0.90
h (head of water at prop center)	10 ft

The propeller analysis method used in this Appendix is similar to the one used in Reference 2.

STEP 1 Determine the ratio K_t/J^2 for both V_{end} and V_{sus} :

IIa $K_t/J^2 = (R)/(N \cdot D^2 \cdot V_{sus}^2 \cdot \rho \cdot (1-w)^2 \cdot (1-t))$

$K_t/J^2 = 0.203$

V-in ft/sec=($V_{kts} \cdot 1.69$)

ρ-1.99

If a 'Series' analysis is used to find EHP for the design speeds, then R can be estimated as follows:

$$R = (EHP \cdot 326)/V_{kts}$$

$K_t/J^2 = (R)/(2 \cdot D^2 \cdot V_{end} \cdot \rho \cdot (1-w)^2 \cdot (1-t))$

$K_t/J^2 = 0.115$

$K_t/J^2 = 0.203 \ldots\ldots V_{sus}$

$K_t/J^2 = 0.115 \ldots\ldots V_{end}$

STEP 2 Make a table of possible K_t and J values for the ratios found in step 1. See Table II-1

30 kts

J	K_t
0.1	.002
0.2	.008
0.3	.018
0.4	.032
0.5	.051
0.6	.073
0.7	.099
0.8	.130
0.9	.164
1.0	.203
1.1	.246
1.2	.293

18 kts

J	K_t
0.1	.001
0.2	.005
0.3	.010
0.4	.018
0.5	.029
0.6	.041
0.7	.056
0.8	.074
0.9	.093
1.0	.115
1.1	.139
1.2	.166

TABLE II-1

STEP 3 Using the values in Table II-1, plot the ratios found in step 1 on a propeller curve like the one in Figure II-1. Where this plot intersects constant P/D lines (points 1,2,3,4 and 5), find the appropriate effeciencies (see red arrows). Connecting these points produces an effeciency curve for that particular propeller operating under those specific inputs. For any point on this curve the thrust, rpm, and effeciency can be determined. As an example take the point of maximum effeciency (point D):

$$\eta_o = 0.7$$

$$J = 1.03$$

$$n = (V_a/J \cdot D) \cdot 60 = (30 \cdot 1.69)/(1.03 \cdot 10.5) \cdot 60$$

$$n = 264.4 \text{ rpm}$$

$$T = K_t \rho \cdot n^2 \cdot D^4 = (.22)(1.99)(4.41)^2(10.5)^4$$

$$T = 103{,}493 \text{ lbs}$$

To insure proper units V_a must be converted to ft/sec in the expression for n and n must be converted to revs/sec in the expression for T.

PROPELLER TYPE = B4-100 : 4 blades; EAR(expanded area ratio) = 1.00

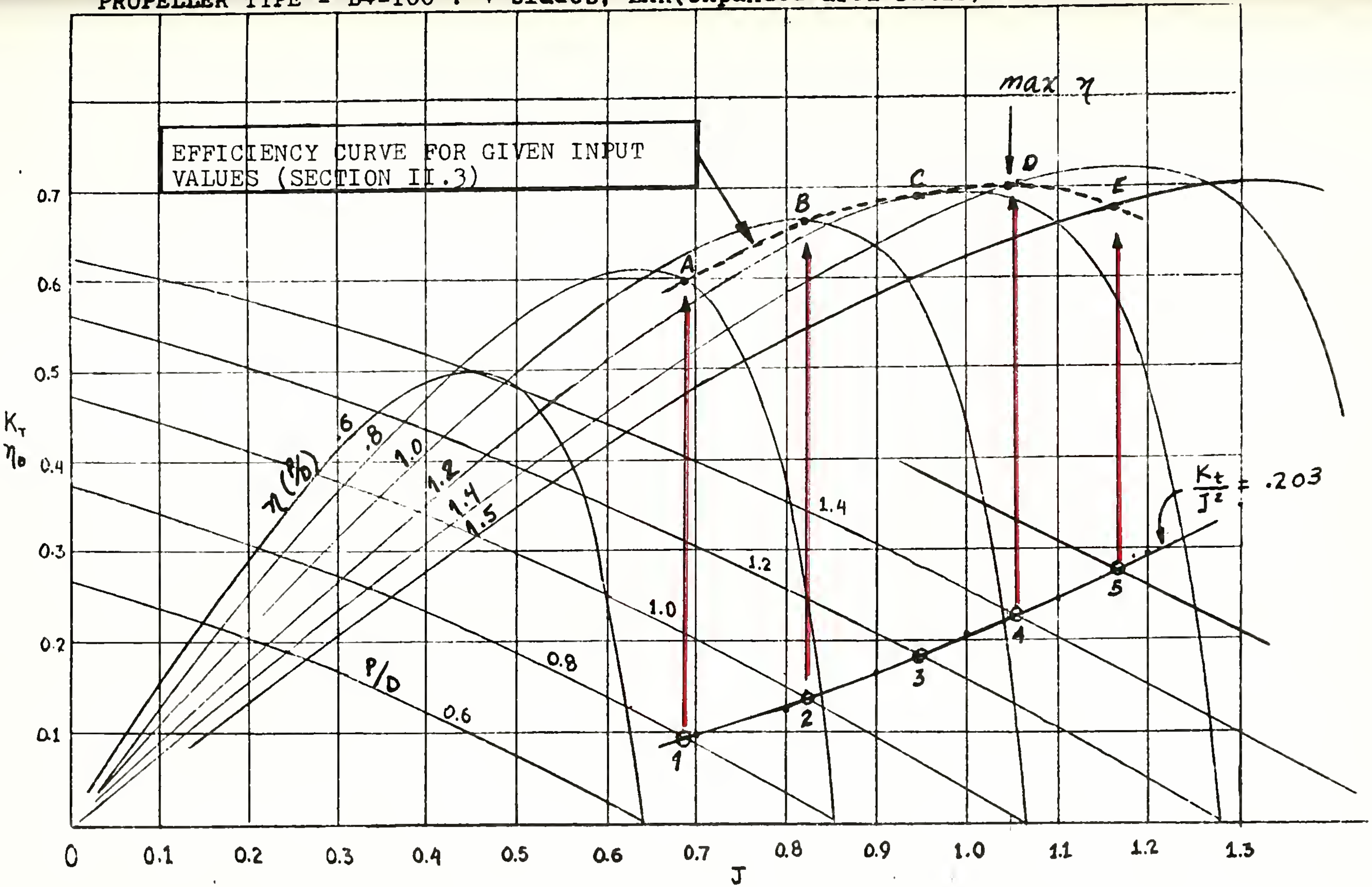


FIGURE II-1 TYPICAL 'TROOST SERIES' PROPELLER CURVE

STEP 4 Determine the cavitation for the propeller (at both speeds) by plotting the thrust loading (τ_c) and the local cavitation number (σ) for a selected point on the propeller curve.* A cavitation diagram, such as the one in Figure II-2, is used in making the plot.

As an example, take the propeller curve in Figure II-1 and calculate values of τ_c and σ using the equations in Figure II-2.

$A_D = (EAR)\frac{\pi D^2}{4} = 86.546$	;propeller curve
$V = 30$ kts	;input
$V_a = (1-w)V = 28.2$ kts	;input
$n = 264.4$	;step 3
$P/D = 1.4$	;Figure II-1
$h = 10$ ft	;input
$D = 10.5$ ft	;input
$t = .10$	;input
$EHP = \frac{RV}{326} = 16{,}711$ hp	;input; step 1

$$q_t = \left(\frac{V_a}{7.12}\right)^2 + \left(\frac{nD}{329}\right)^2 = \left(\frac{28.2}{7.12}\right)^2 + \left(\frac{264.4\ 10.5}{329}\right)^2$$

$$q_t = 86.89 \text{ psi}$$

$$p_o - p_v = 14.45 + 0.45h = 14.45 + 0.45(10)$$

$$p_o - p_v = 18.95 \text{ psi}$$

* Usually the maximum effeciency or a design rpm are the points of most interest.

$$A_p = A_D(1.067-0.229(P/D))$$

$$A_p = (86.546)(1.067-0.229(1.4))$$

$$A_p = 64.90$$

$$T/A_p = \frac{2.26EHP(1+x)}{(1-t)VA_p} = \frac{2.26(16,711)(1.0004)}{(.9)(30)(64.9)}$$

$$T/A_p = 21.56 \text{ psi}$$

$$\tau_c = \frac{T}{A_p q_t} = (21.56)\left(\frac{1}{86.89}\right)$$

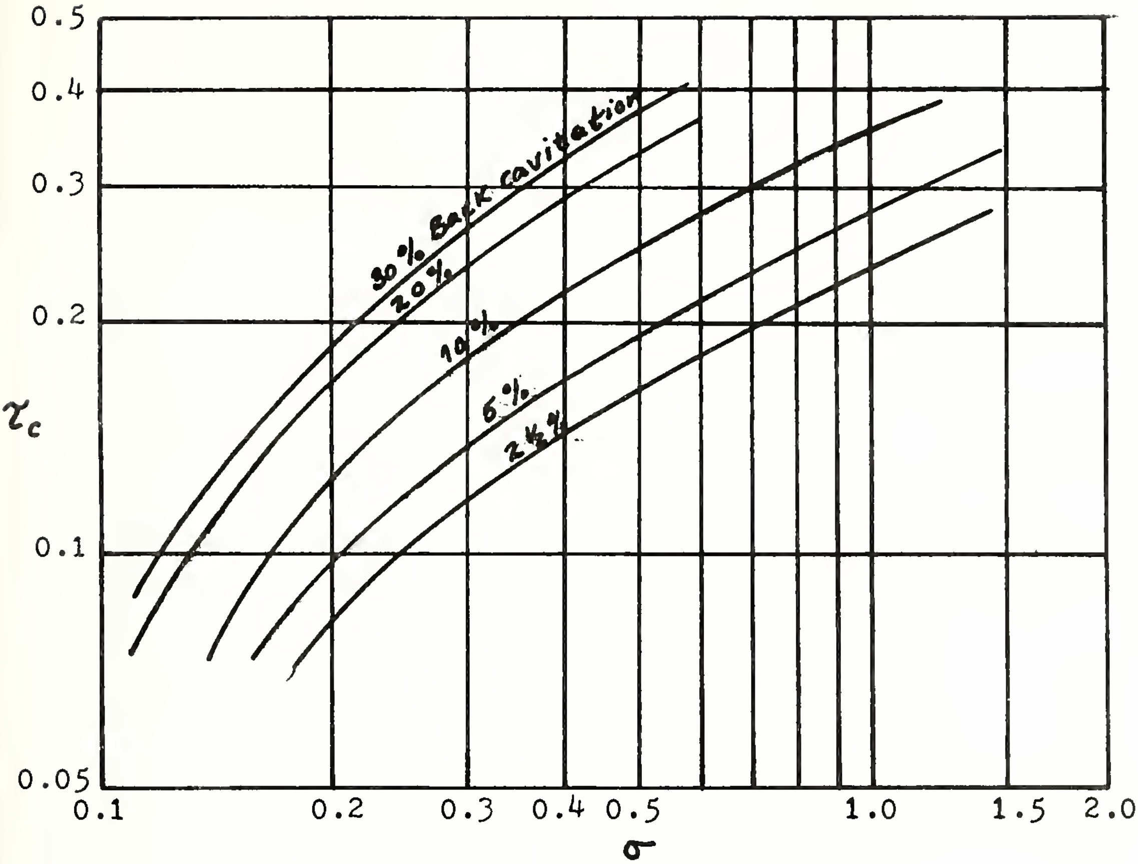
$$\tau_c = .25$$

$$\sigma = \frac{p_o - p_v}{q_t} = \frac{18.95}{86.89}$$

$$\sigma = .22$$

Plotting τ_c and σ on Figure II-2 shows that there will be about 40% back cavitation for this propeller at 30 kts.

step 5 Repeat steps 1 through 5 for different propellers and construct a propeller trade-off table similar to Table II-2. From this table a propeller selection can be made, using the guide lines presented in Section 2.5.2



$$\tau_c = T/(A_p \cdot q_t) \quad ; \quad \sigma = \frac{p_o - p_v}{q_t}$$

$$T/A_p = (2.26)\,\frac{EHP\ (1+x)}{(1-t)\cdot V \cdot A_p} \quad ; \quad psi$$

$$q_t = \left(\frac{V_a}{7.12}\right)^2 + \left(\frac{nD}{329}\right)^2 \quad ; \quad psi$$

$$p_o - p_v = 14.45 + 0.45h$$

$$A_p = A_D\ (1.067 - 0.229(P/D))$$

1+x - model ship correlation factor (usually 1.0004)

A_D - $\frac{(EAR)\, D^2 \pi}{4}$

h - head of water at screw centerline

V and V_a are in kts

n is in rpm

D is in ft

FIGURE II-2 SIMPLE CAVITATION DIAGRAM (2)

Prop	$\bar{V}$	$\frac{P}{d}$	J	n	q_t	A_p	$\frac{T}{A_p}$	$\frac{T}{A_p q_t}$	$\sigma_{.7R}$	% cav	η_o
B3-35	30	1.00	.80	340.42	133.72	25.40	27.59	.210	.140	30	.730
B3-35	18	1.40	1.15	142.08	126.21	22.62	2.80	.110	.720	0	.770
B3-50	30	1.12	.85	320.39	120.24	35.15	19.94	.170	.160	30	.710
B3-50	18	1.40	1.14	143.33	26.57	32.32	1.96	.070	.710	0	.760
B3-65	30	1.10	.85	320.39	120.24	45.87	15.27	.130	.160	18	.690
B3-65	18	1.40	1.14	143.33	26.57	42.01	1.51	.060	.710	0	.740
B3-80	30	1.40	1.01	269.64	89.74	51.70	13.55	.150	.210	17	.680
B3-80	18	1.40	1.13	143.97	26.76	51.70	1.22	.050	.710	0	.730
B4-40	30	1.40	1.04	261.87	85.54	25.86	27.09	.320	.220	30	.720
B4-40	18	1.40	1.18	138.48	25.18	15.86	2.54	.100	.750	0	.755
B4-55	30	1.35	1.01	269.40	89.53	36.09	19.41	.217	.211	30	.715
B4-55	18	1.40	1.17	138.98	25.30	35.54	4.02	.158	.749	0	.760
B4-70	30	1.40	1.04	262.88	86.00	45.24	15.48	.180	.220	20	.705
B4-70	18	1.40	1.16	140.77	25.81	45.24	3.16	.122	.734	0	.750
B4-85	30	1.41	1.04	261.62	85.33	54.76	12.79	.150	.222	12	.700
B4-85	18	1.39	1.15	141.38	25.98	55.10	2.59	.099	.729	0	.735
B4-100	30	1.40	1.03	264.40	86.89	64.90	21.56	.250	.220	40	.700
B4-100	18	1.44	1.16	140.77	25.81	63.83	2.24	.086	.734	0	.710
B5-45	30	1.39	1.05	257.90	83.36	29.17	24.01	.028	.227	30	.720
B5-45	18	1.40	1.19	137.22	24.80	29.08	4.91	.197	.764	2	.740
B5-60	30	1.40	1.05	257.90	83.36	38.78	18.06	.216	.227	30	.720
B5-60	18	1.40	1.19	137.22	24.80	38.78	3.68	.148	.764	0	.750
B5-75	30	1.24	.96	283.43	97.42	50.85	13.77	1.41	1.94	17	.715
B5-75	18	1.46	1.22	138.85	25.26	47.58	3.00	1.18	.750	2	.750

TABLE II-2

APPENDIX III - GEAR SIZING

III.1 <u>'K' FACTOR</u> One of the most controlling design elements in reduction gear design is the amount of stress at the point of tooth contact: The 'K' factor is a measure of this tooth surface stress (8).

$$\text{IIIa} \qquad K = (R+1/R)(W_t/F_e \cdot d) \; : \qquad \frac{\text{loading}}{\text{in face-in dia}}$$

d - pitch dia of pinion (in)
R - gear ratio
W_t- total tangential tooth load (lb)
F_e- effective face width (in)

For Navy ships 'K' factor values vary between 110 and 150. This range insures that stresses do not promote excessive wear and the gear is not over designed for the applied torque

III.2 <u>GEAR SIZING</u> The 'K' factor is related to the various design parameters in the following manner (8).

$$\text{IIIb} \qquad K = 126{,}050 \cdot SHP \cdot (R+1)/N_p \cdot d^2 \cdot F_e \cdot R$$

N_p- pinion revolutions per minute

Good design practices, aimed at avoiding excessive deflections, limits the face-width to diameter ratio to the range 2.0-2.25 : Pinion gear (8)

$$\text{IIIc} \qquad 2.0 \leq F_e/d \leq 2.25$$

Combining expressions IIIb and IIIc gives an expression useful in determining gear diameter:

$$\text{IIId} \qquad d^3 = 126{,}050 \cdot SHP \cdot (R+1)/N_p \cdot 2.25 \cdot R \cdot K$$

III.3 <u>GEAR WEIGHT</u> Probably more important than gear size is gear weight (assuming reduction gear designers stay within emperical sizing lanes). For preliminary design purposes, an emperical relationship by Dudley gives excellent results (13).

$$\text{IIIe} \qquad W = A(Q/K)^n$$

Q - $SHP(R+1)^3/N_p \cdot R^*$: SHP of an input pinion
A - emperical constant
4.25-planetary gears
12.95-conventional gears
n - emperical constant: $.8 \leq n \leq 1.0$

* - For double reduction gears:
$R_1 = (2R_1)$ and $R_2 = R_2$

III.4 <u>EXAMPLE GEAR CALCULATION</u> The gear diameter and weight are the two most critical design characteristics (with respect to reduction gears), and can be calculated quite easily using equations IIId and IIIe. Assume, for instance, we are sizing a single input, double reduction, locked-train reduction gear . This gear will be used to couple a gas turbine to a CRP propeller, designed for 150 rpm at maximum speed.

INPUTS

Prime Mover	LM2500
Horsepower	22,000
RPM (Prime Mover)	3,600
'K' factor (high speed pinion)	140
'K' factor (low speed pinion)	110
Propeller RPM	150

STEP 1 Determine R_1 and R_2 (first and second reduction ratios). A useful approximation for R_2 is:

$$R_2=\sqrt{N_{in}/N_{Prop}} - 1$$; conventional gear without locked train

$$R_2=\sqrt{N_{in}/N_{Prop}} + 3$$; locked train

For this example then:

$$R_2=\sqrt{3600/150} + 3 = 7.9$$

$R_1 = 3600/150 \cdot 7.9 = 3.04$

STEP 2 Calculate the diameter of the first reduction pinion using equation IIId.

$$d_1^3 = 126{,}050 \cdot 11{,}000 \overset{*}{\cdot} 4.04/3600 \cdot 2.25 \cdot 3.04 \cdot 140$$

$$d_1 = 11.7 \text{ in}$$

* For a locked train, the input SHP gets split between the two first reduction pinions

STEP 3 Compute the diameter of the first reduction gear.

$$D_1 = R_1 \cdot d_1$$

$$D_1 = 35.56 \text{ in}$$

Step 4 Calculate the diameter of the second reduction pinion.

$$d_2^3 = 126{,}050 \cdot 11{,}000 \cdot 8.9/(150 \cdot 7.9)\ 2.25 \cdot 7.9 \cdot 110$$

$$d_2 = 10.78 \text{ in}$$

STEP 5 Calculate the diameter of the second reduction gear (bull gear).

$$D_2 = R_2 \cdot d_2$$

$$D_2 = 85.19 \text{ in } (7.10 \text{ ft})$$

STEP 6 Calculate the gear weights: IIIe

$$W_1 = 12.95(22{,}000)(4.04)^3/3600(2x3.04)(140)$$

$$W_1 = 6.13 \text{ tons } \times 2 \text{ (1st reduction gears)} = 12.26$$

$$W_2 = 12.95\underline{(22{,}000)}_{2}(8.9)^3/(150 \cdot 7.9)(7.9)(110)$$

$$W_2 = 97.5 \text{ tons}$$

$$W_{total} = 109.76 \text{ tons}$$

With these gear diameters and weights the reduction gear can be sized easily.These values represent the most restricting weights and dimensions of the reduction gear assembly. Gear width, casing weight and other aspects of reduction gear sizing can also be estimated using information found in Reference (8).

APPENDIX IV - PROPULSION PLANT SIZING CURVES

IV.1 <u>SIZING CURVES</u> The figures in this appendix allow various Group 2 weights and volumes to be determined, provided the installed shaft horsepower is known. For KW_I an estimate of displacement is necessary. References 1, 3, 5, 7, 9 and 10 were used in deriving the figures.

IV.2 <u>LIST OF FIGURES</u>

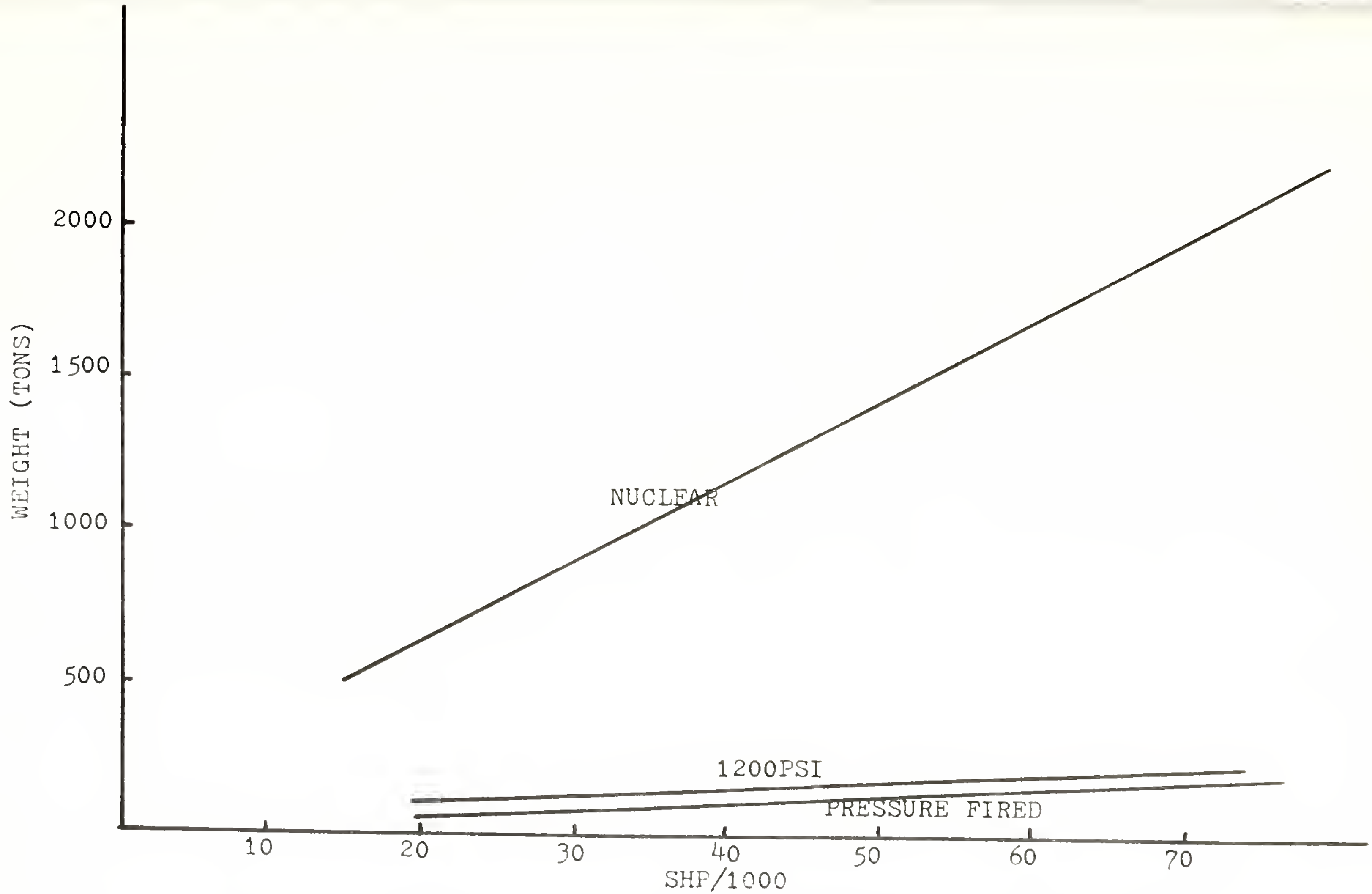


FIGURE IV-1 Weight Group 200 (energy converters) vs Horsepower

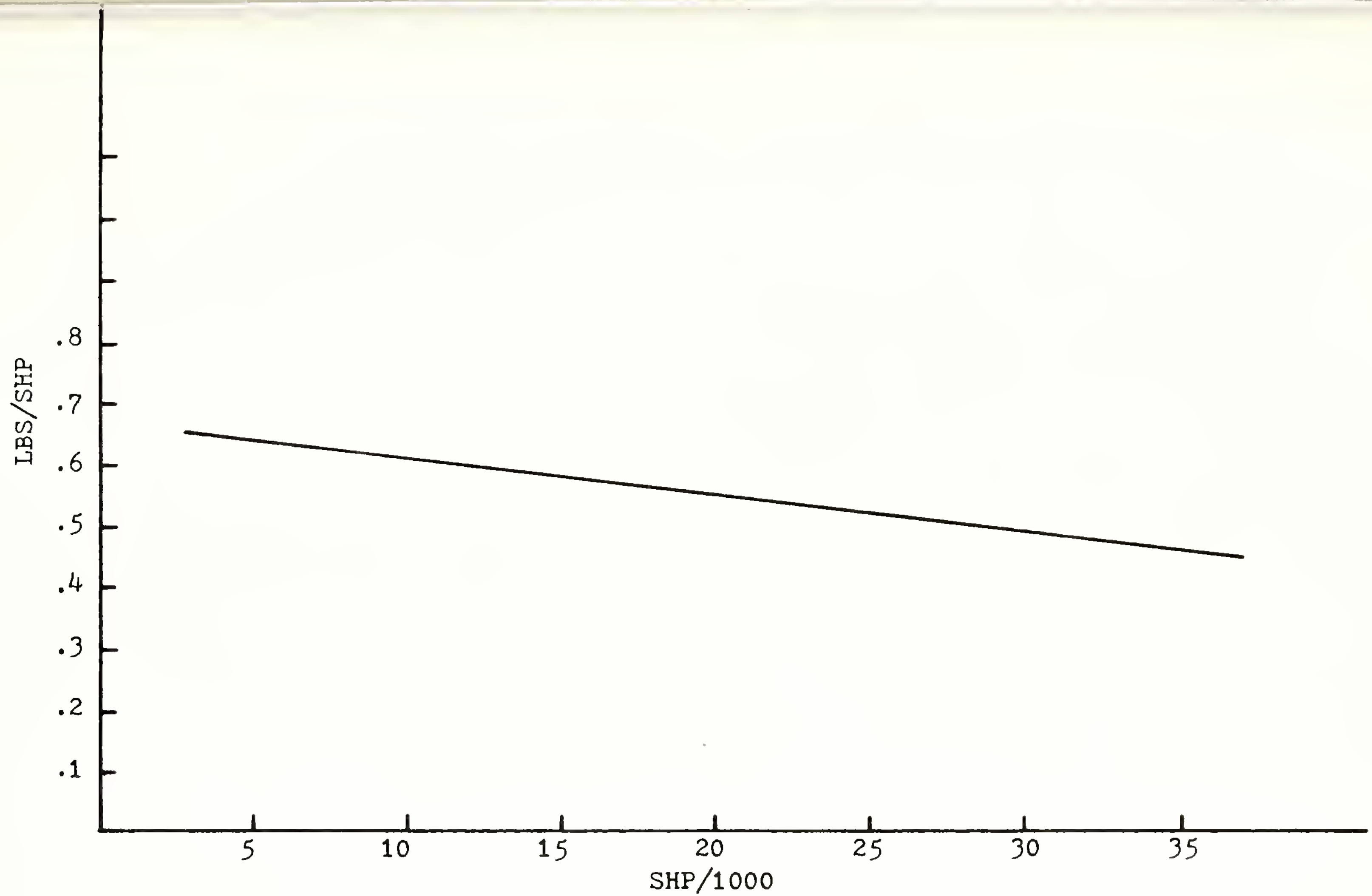


FIGURE IV-2 Gas Turbines: Specific Weight vs Horsepower

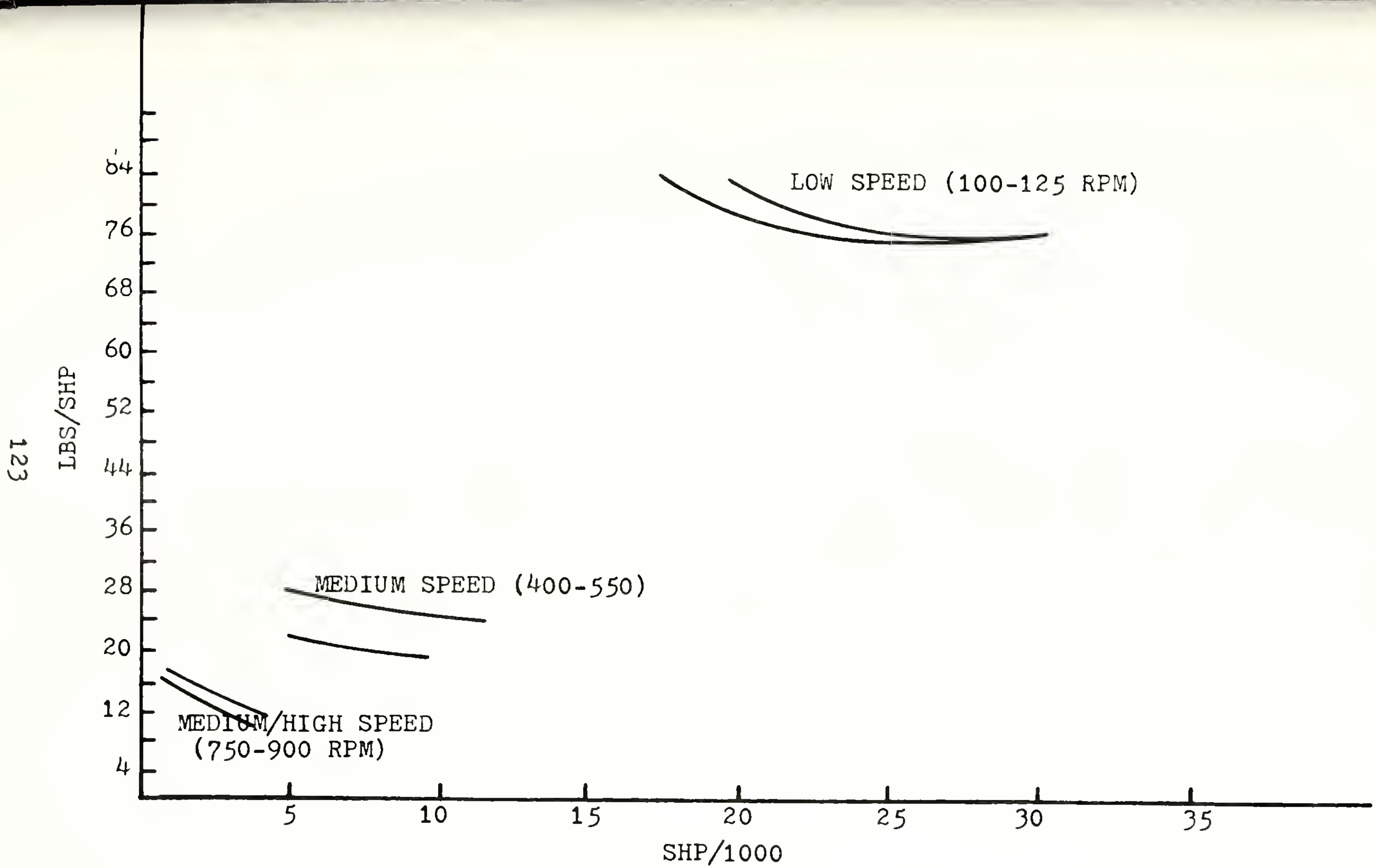


FIGURE IV-3 Diesel Engines: Specific Weight vs Horsepower

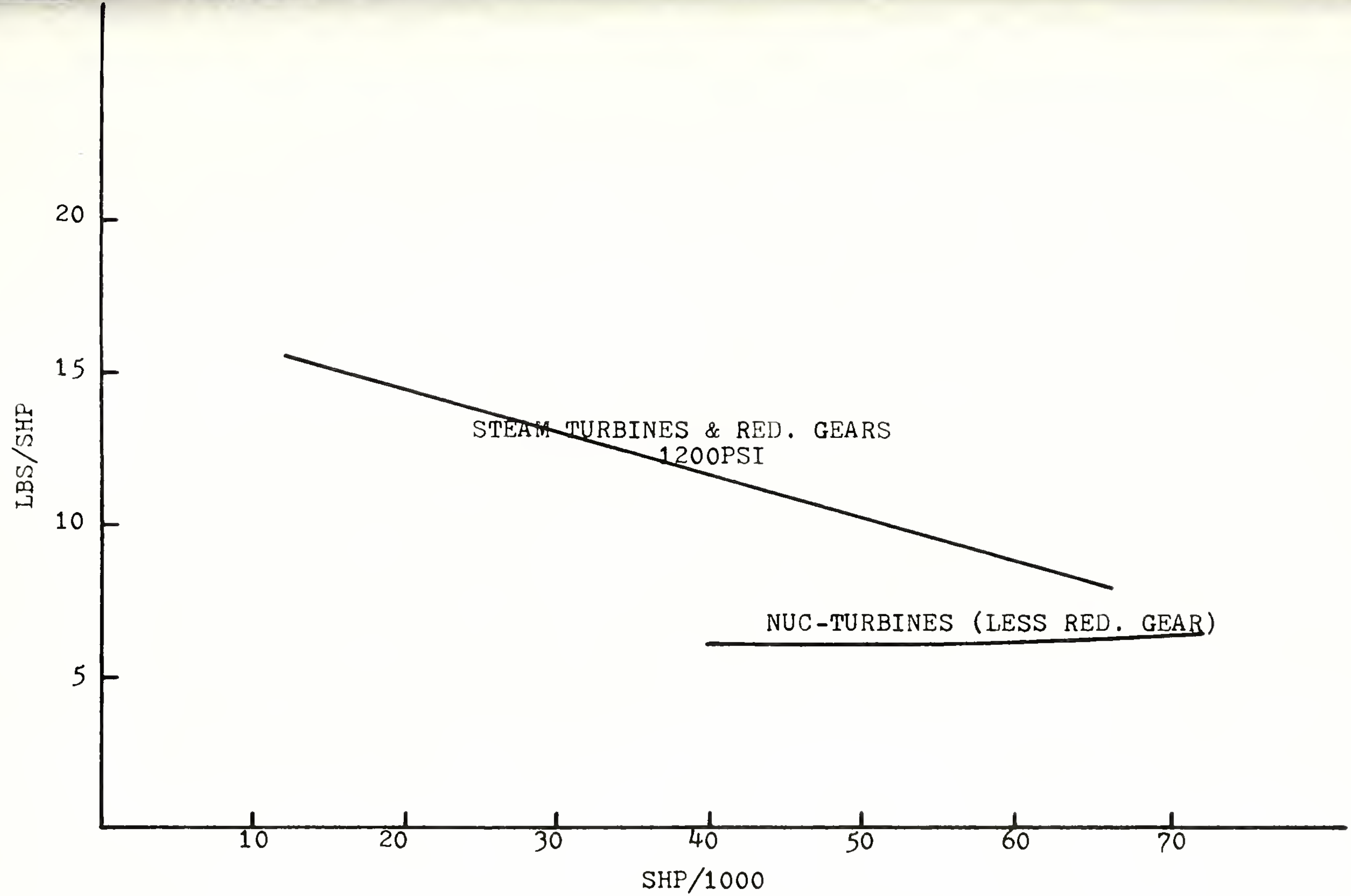


FIGURE IV-4 Steam Turbines: Specific Weight vs Horsepower

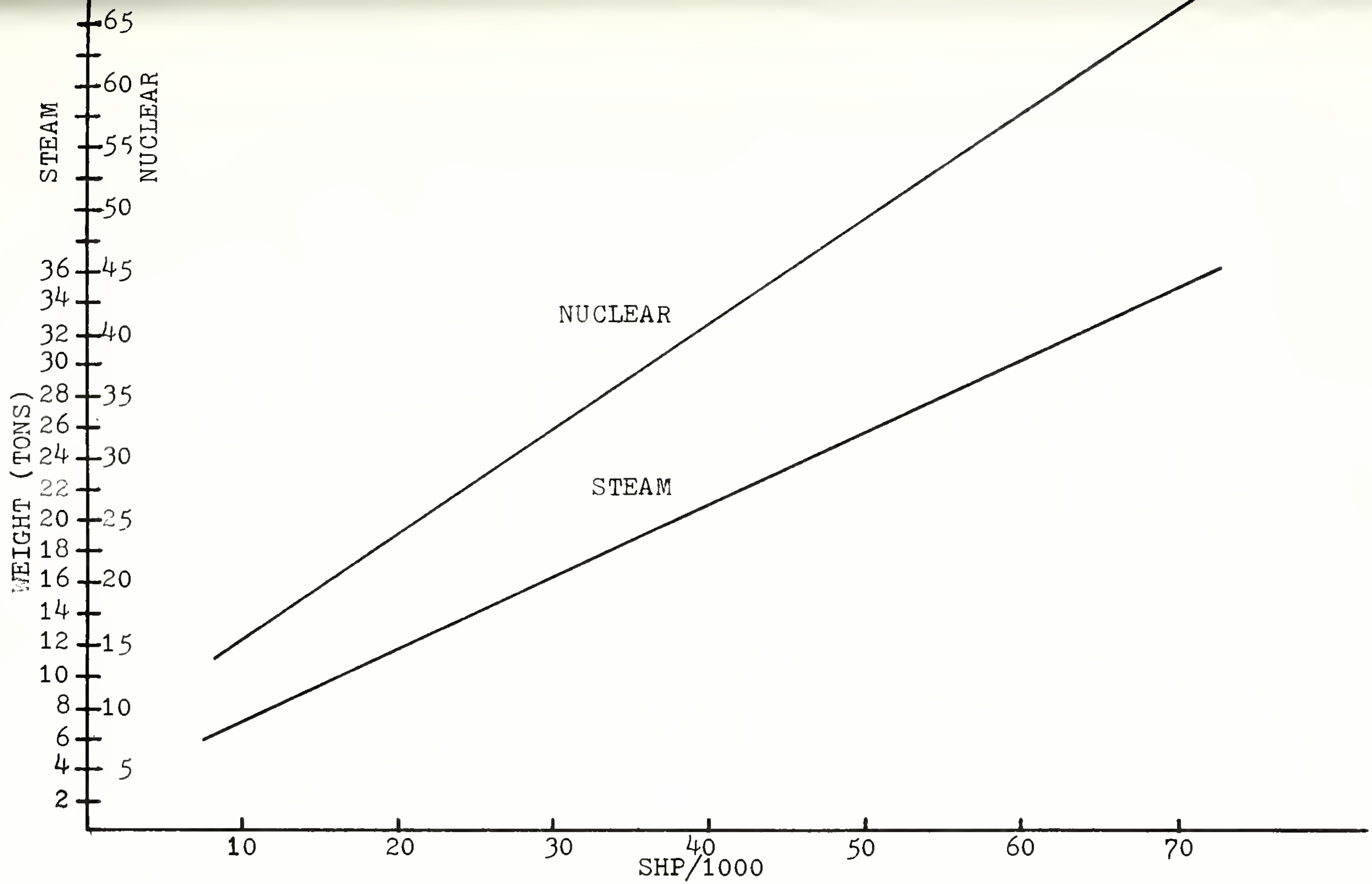


FIGURE IV-5 Main Condenser&Air Ejectors Weight vs Horsepower

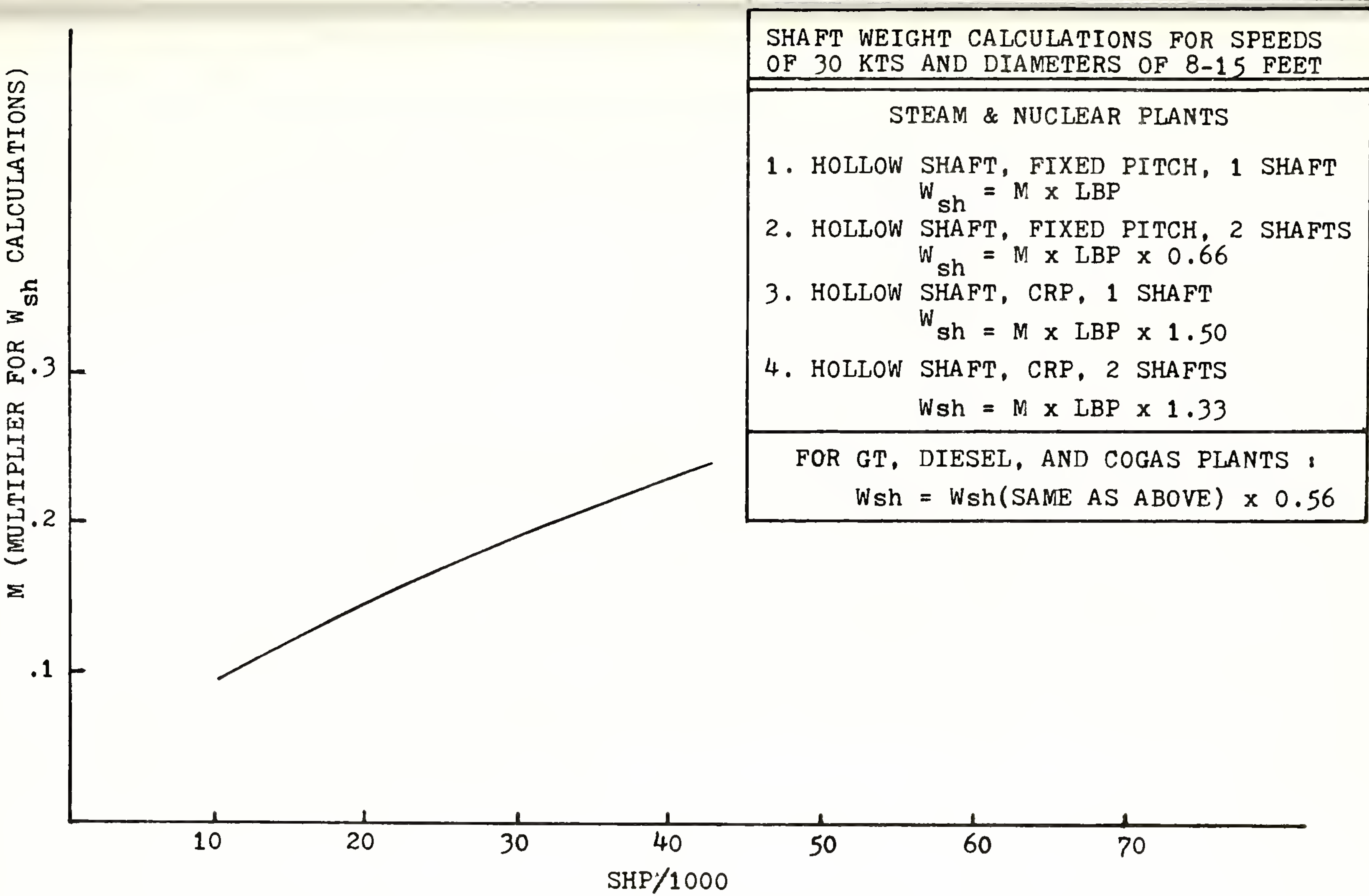


FIGURE IV-6 Shaft Weight vs Horsepower

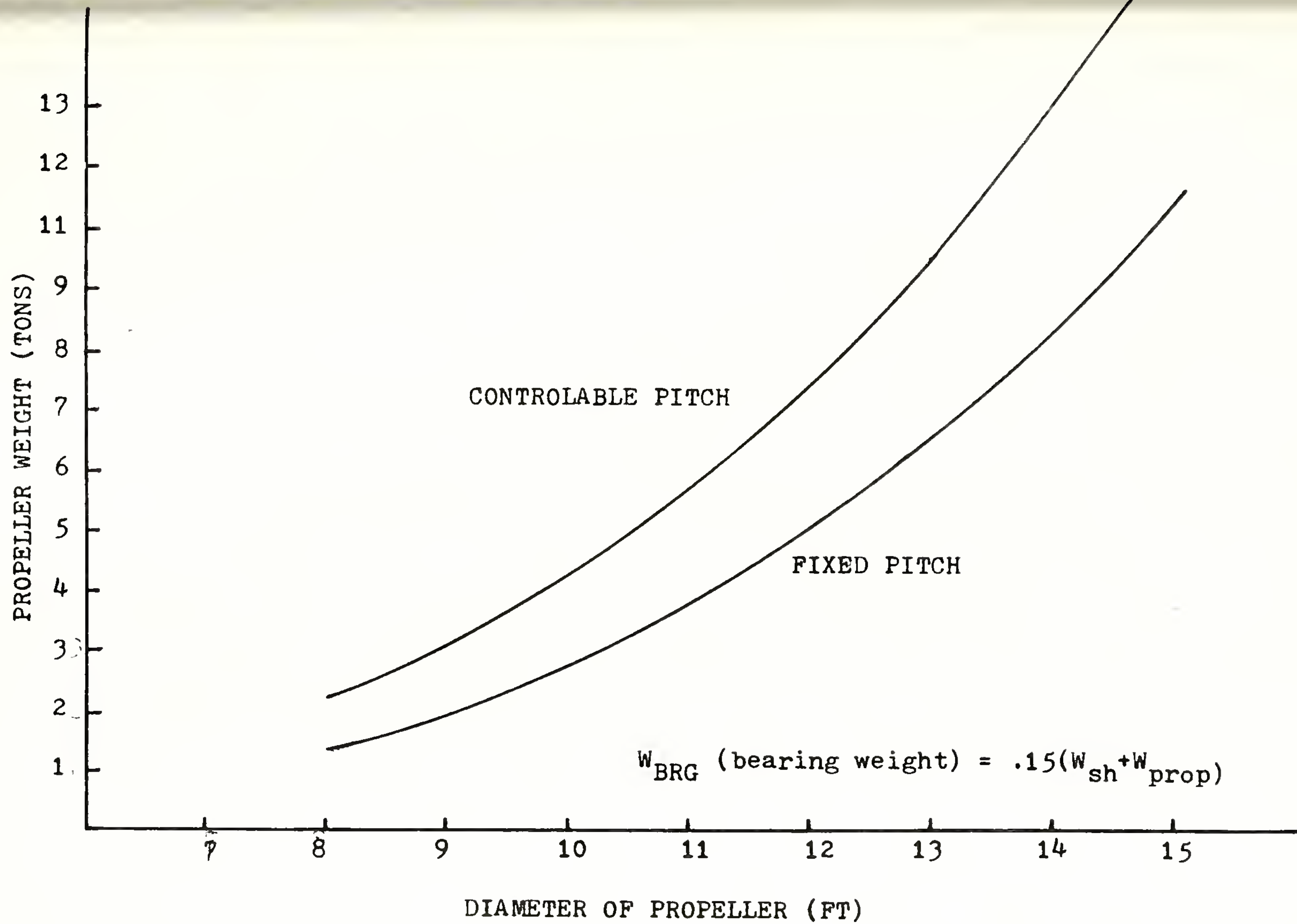


FIGURE IV-7 Propeller Weight vs Propeller Diameter, and Weight of Shaft Bearings

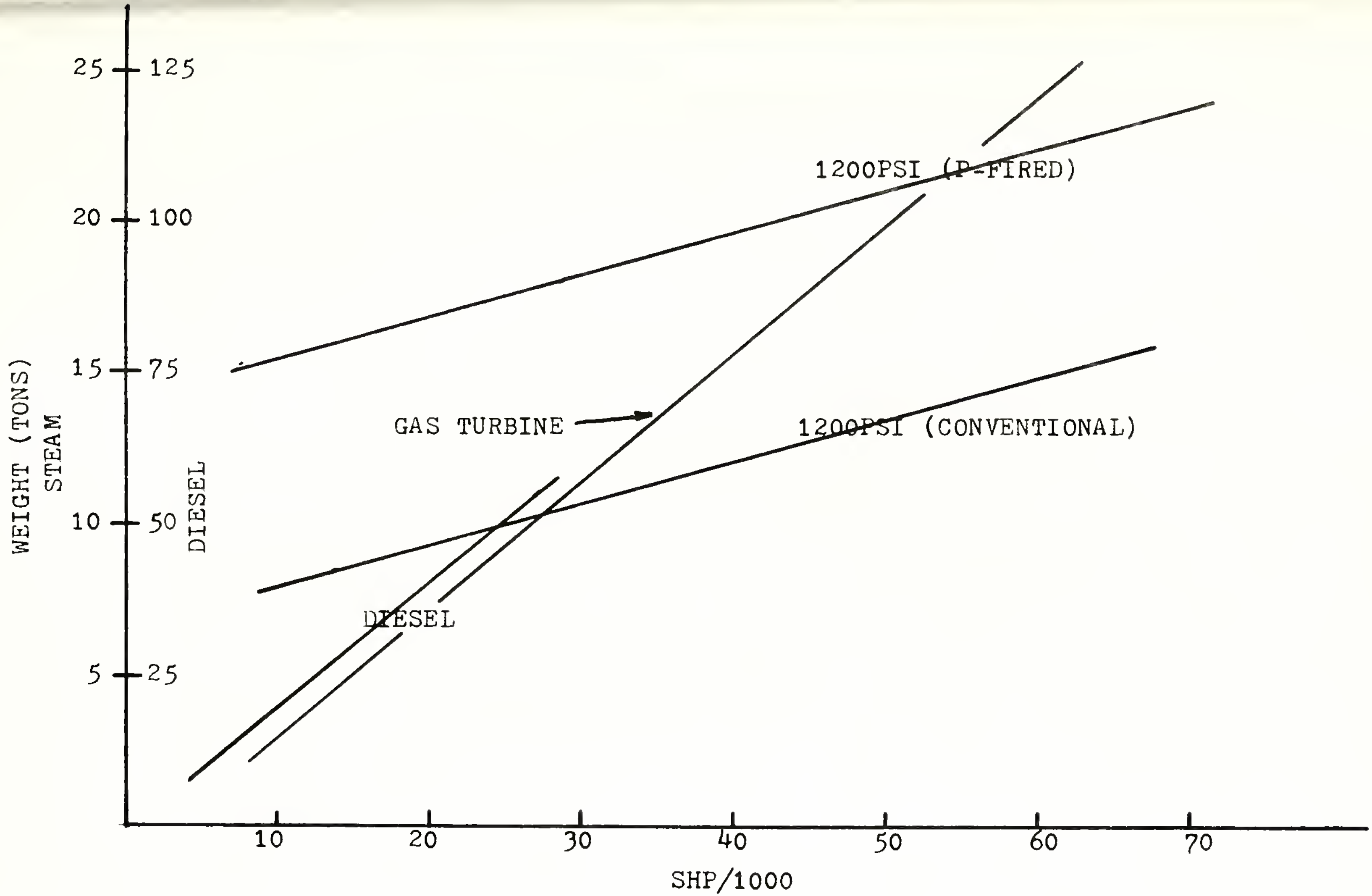


FIGURE IV-8 Weight of Uptakes vs Horsepower

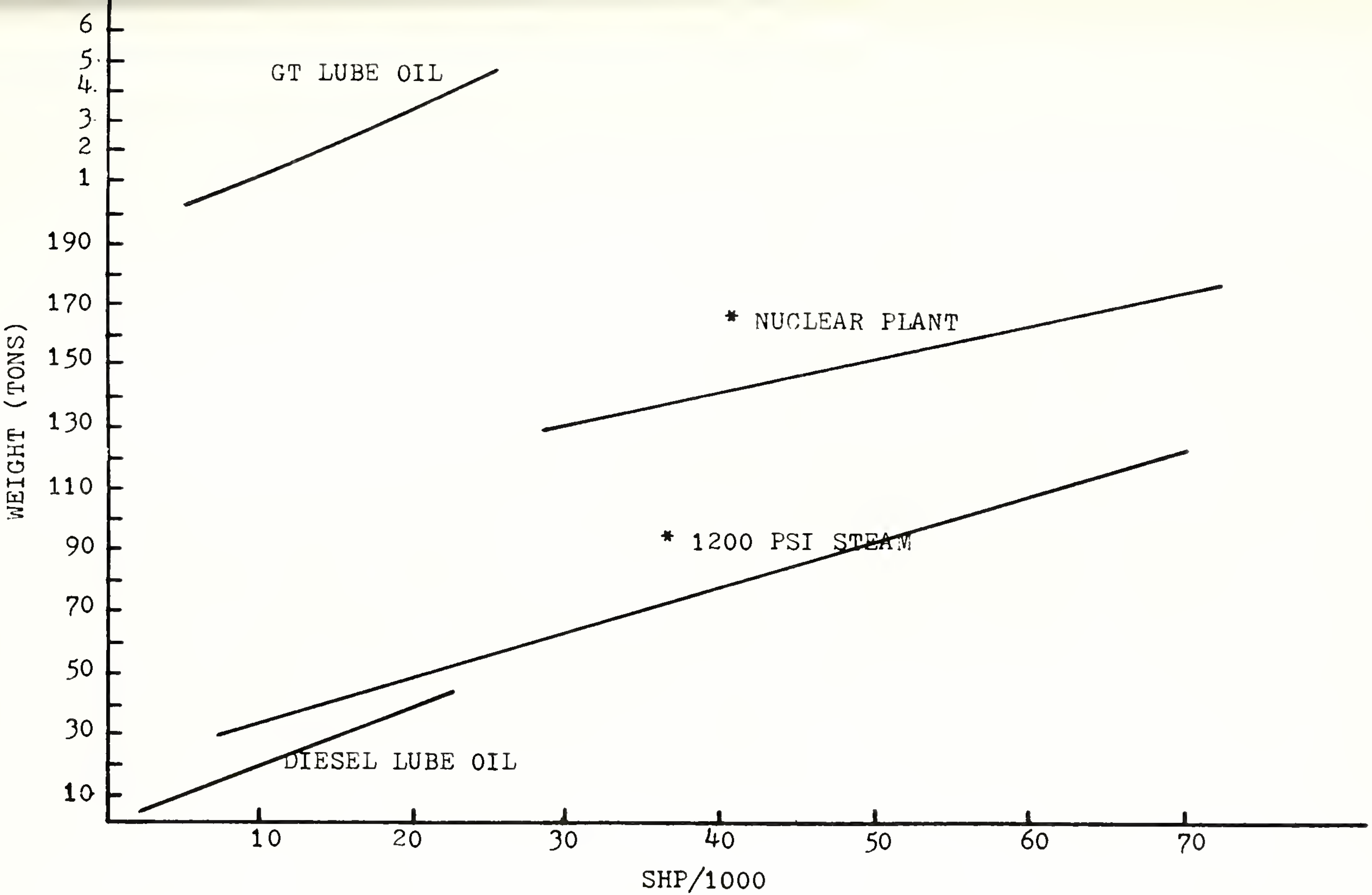


FIGURE IV-9 *Steam, Feed&Condensate, Circ Water and Lube Oil vs Horsepower

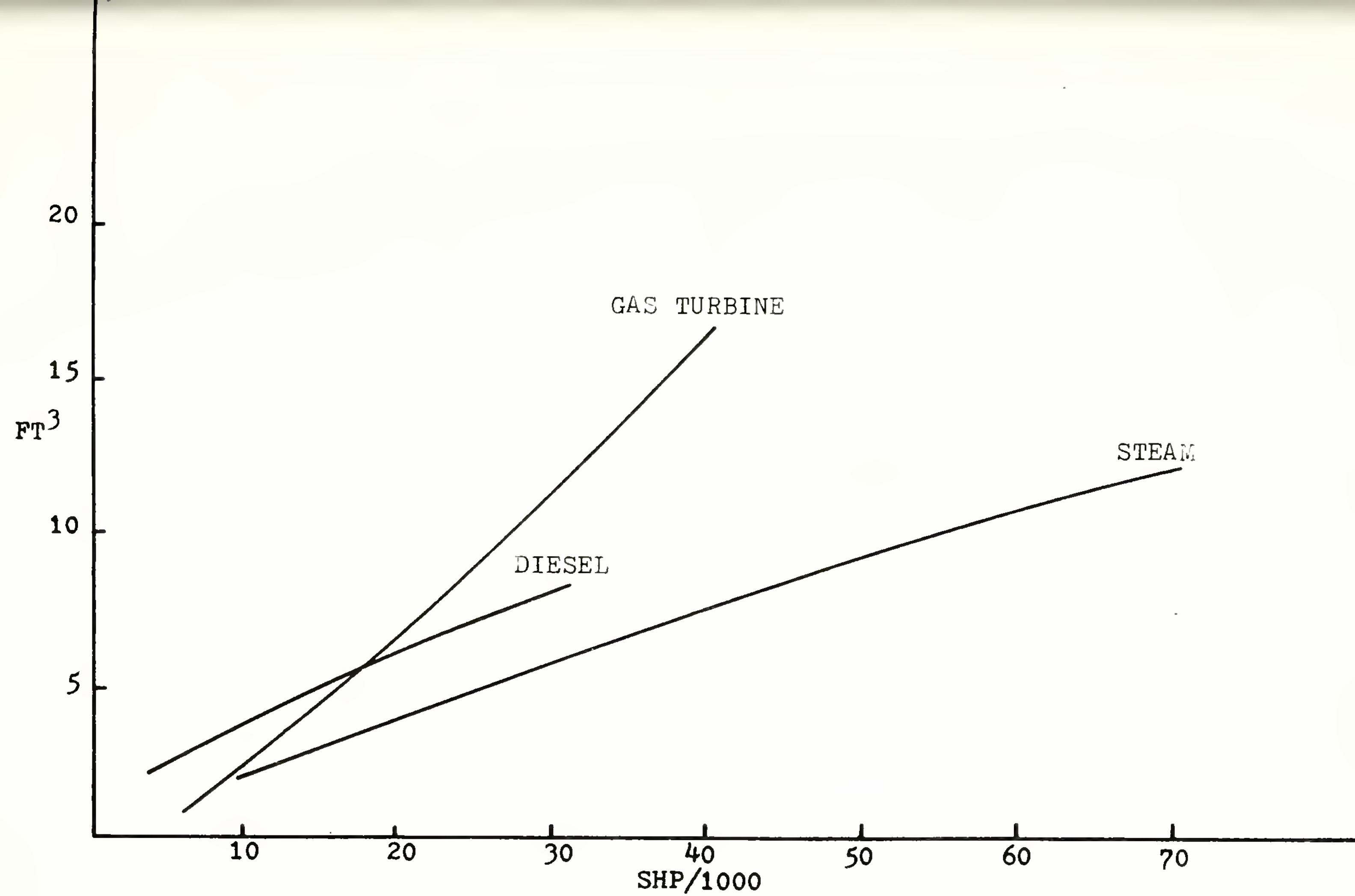


FIGURE IV-10 Uptakes Volume vs Horsepower

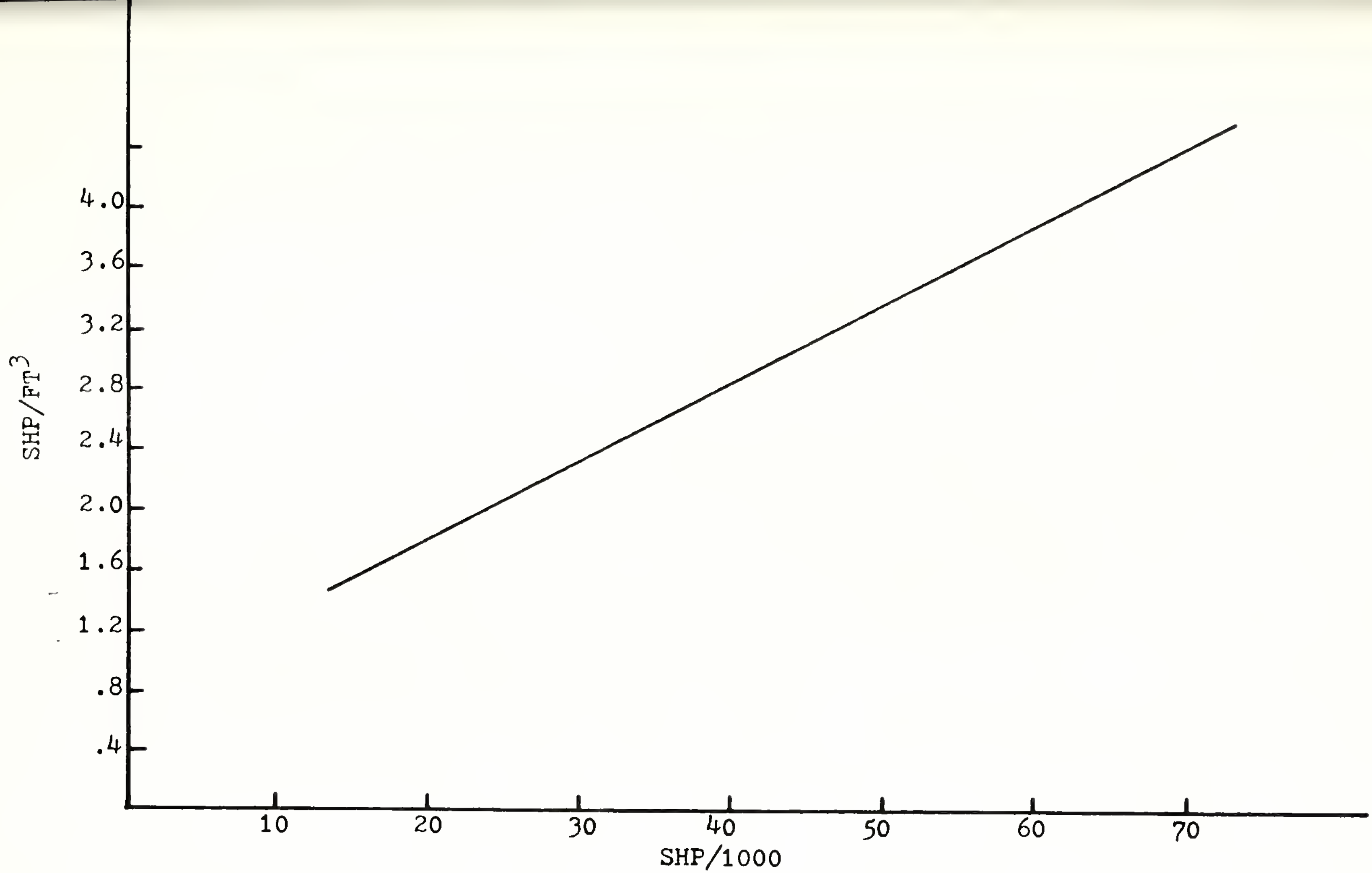


FIGURE IV-11 Steam Turbines (including reduction gears): Power Density vs Horsepower

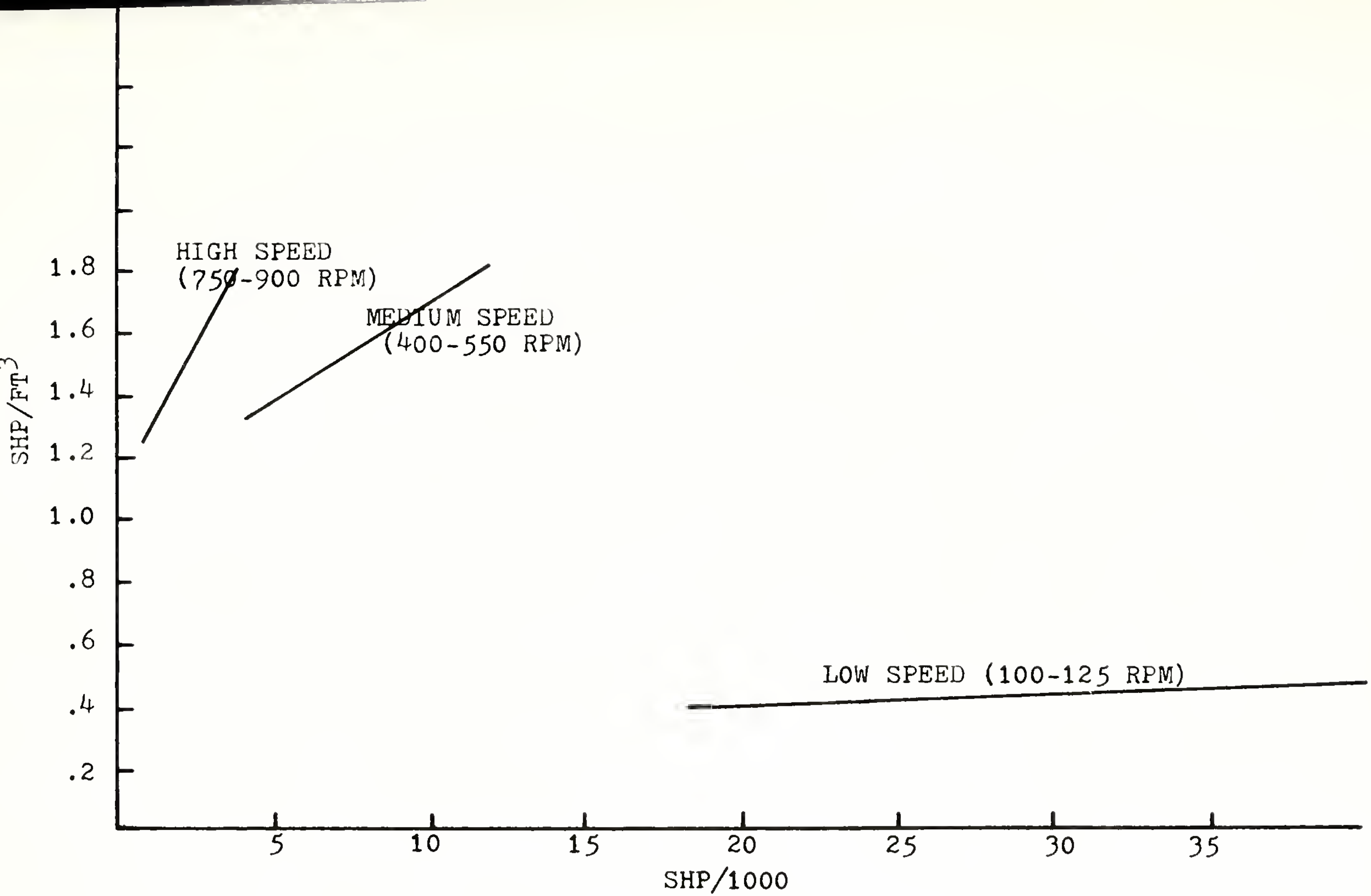


FIGURE IV-12 Diesel Engine: Power Density vs Horsepower

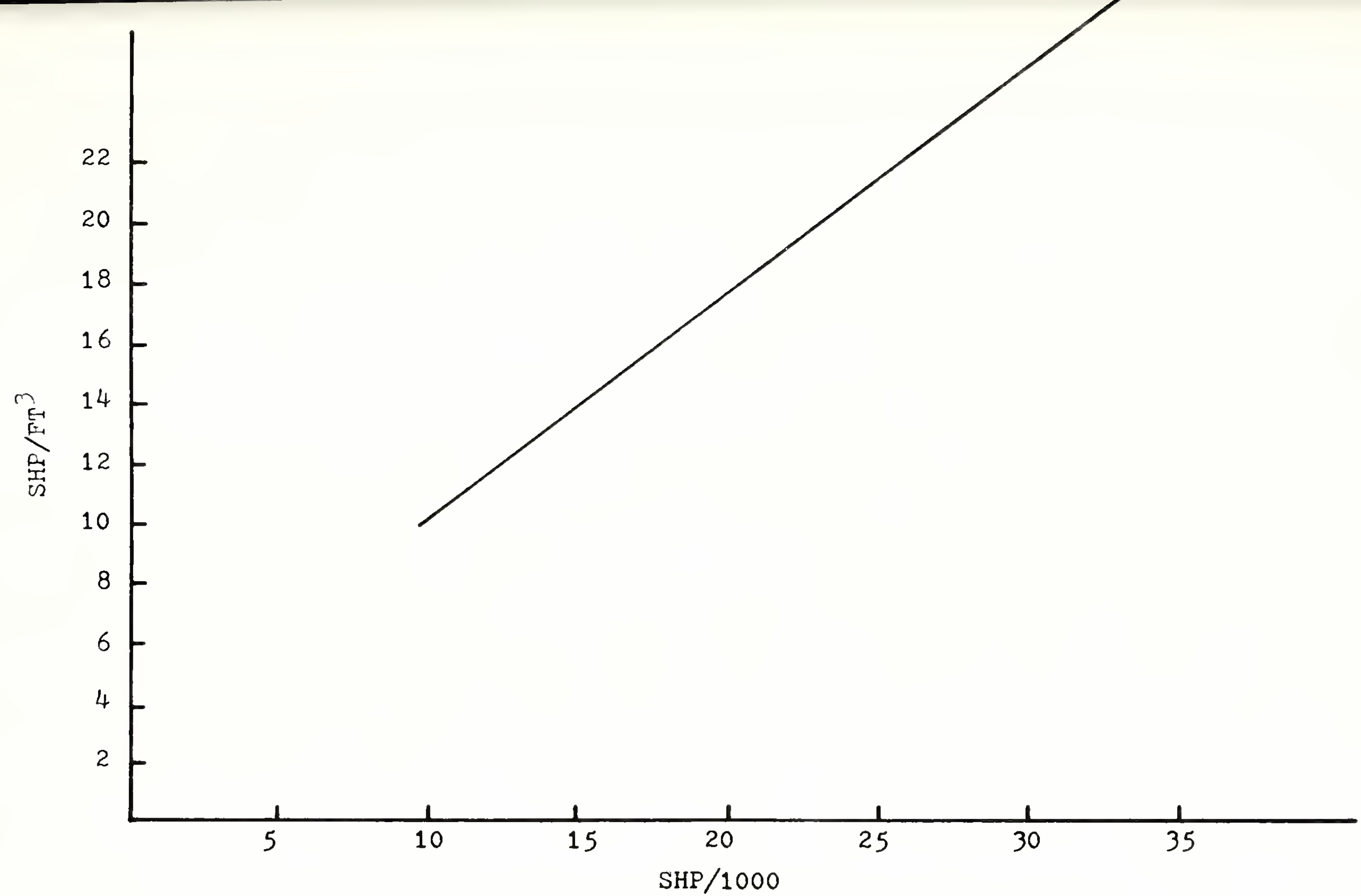


FIGURE IV-13 Gas Turbines: Power Density vs Horsepower

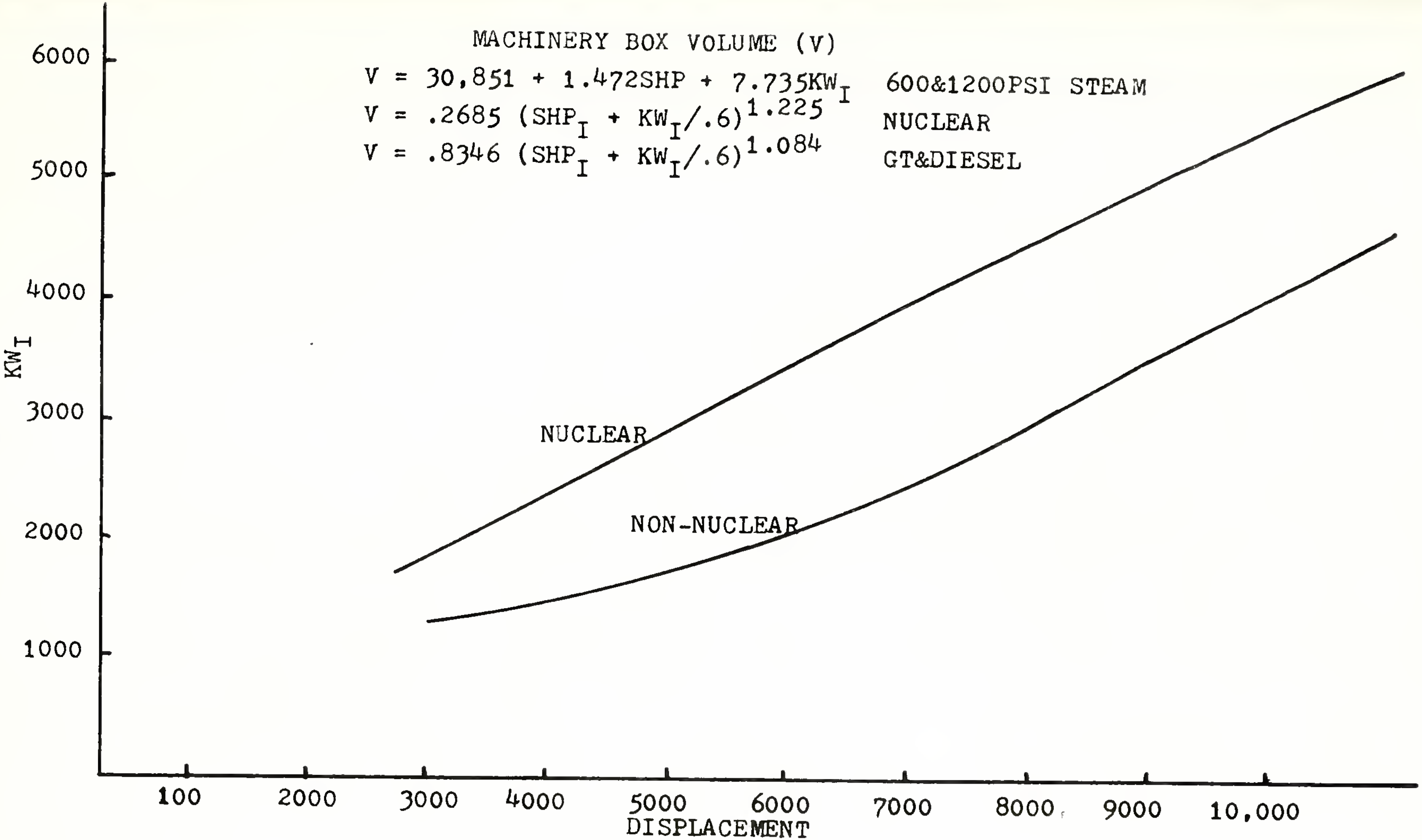


FIGURE IV-14 Installed KW vs Displacement

APPENDIX V - PROPULSION PLANT MACHINERY DATA

V.1 INTRODUCTION This appendix presents useful performance data on basic propulsion components. The curves on boiler and diesel SFC's are taken from Reference 1.

V.2 LIST OF FIGURES AND TABLES

The Troost curves were reproduced from MIT library copies.

MODEL	POWER RATING				RPM	WEIGHT
	NORMAL		MAXIMUM			
	(hp)	(sfc)	(hp)	(sfc)		
GE LM2500	20,000	0.41	27,000	0.39	3,600	11,600
GE LM1500	12,500	0.58	14,000	0.57	5,500	7,500
GE LM100	1,000	0.65	NA	NA	19,500	350
TGP/GTPF 990	5,000	0.46	6,200	0.44	3,600	3,201
TP&MS FT4A-2	24,200	0.50	30,000	0.48	3,600	14,200
TP&MS FT4A-12	26,950	0.51	31,500	0.51	3,600	14,200
TP&MS FT4A-14	31,150	0.52	34,700	0.52	3,600	14,300
TP&MS FT4C-2	35,500	0.47	44,100	0.46	3,600	NA
TP&MS FT12A-3	2,500	0.82	3,770	0.72	9,000	1,150
TP&MS FT12A-6	3,150	0.74	4,180	0.71	9,000	1,150
Allison 501K	3,780	0.54	NA	NA	13,820	2,500
Solar T3001	3,000	0.56	3,120	0.55	14,300	5,500
Lycoming TF12A	1,000	0.72	1,100	0.62	18,500	920
Lycoming TF14B	1,250	0.60	1,375	0.59	18,500	920
Lycoming TF25A	2,000	0.63	2,200	0.62	14,000	1,020
Lycoming TF35	2,500	0.57	2,750	0.56	14,000	1,090

TABLE V-1 GAS TURBINE MARINE ENGINES (11)

	BHP	RPM	SFC	L.O. CONSUMP. bhp-hr/gal
GM				
8-645E5	1,450	900	0.38	3,410
12-645E5	2,150	900	0.37	3,410
16-645E5	2,875	900	0.378	3,410
20-645E5	3,600	900	0.375	3,410
FAIRBANKS				
6 CYL	1,800	900	0.375	4,000
9 CYL	2,700	900	0.375	4,000
12-CYL	3,600	900	0.375	4,000
ENTERPRISE				
R-46	3,656	450	0.369	15,000
R-48	4,875	450	0.369	15,000
RV-12-4	7,313	450	0.369	15,000
RV-16-4	9,751	450	0.369	15,000
RV-20-4	12,187	450	0.369	15,000

TABLE V-2 DIESEL ENGINES (1)

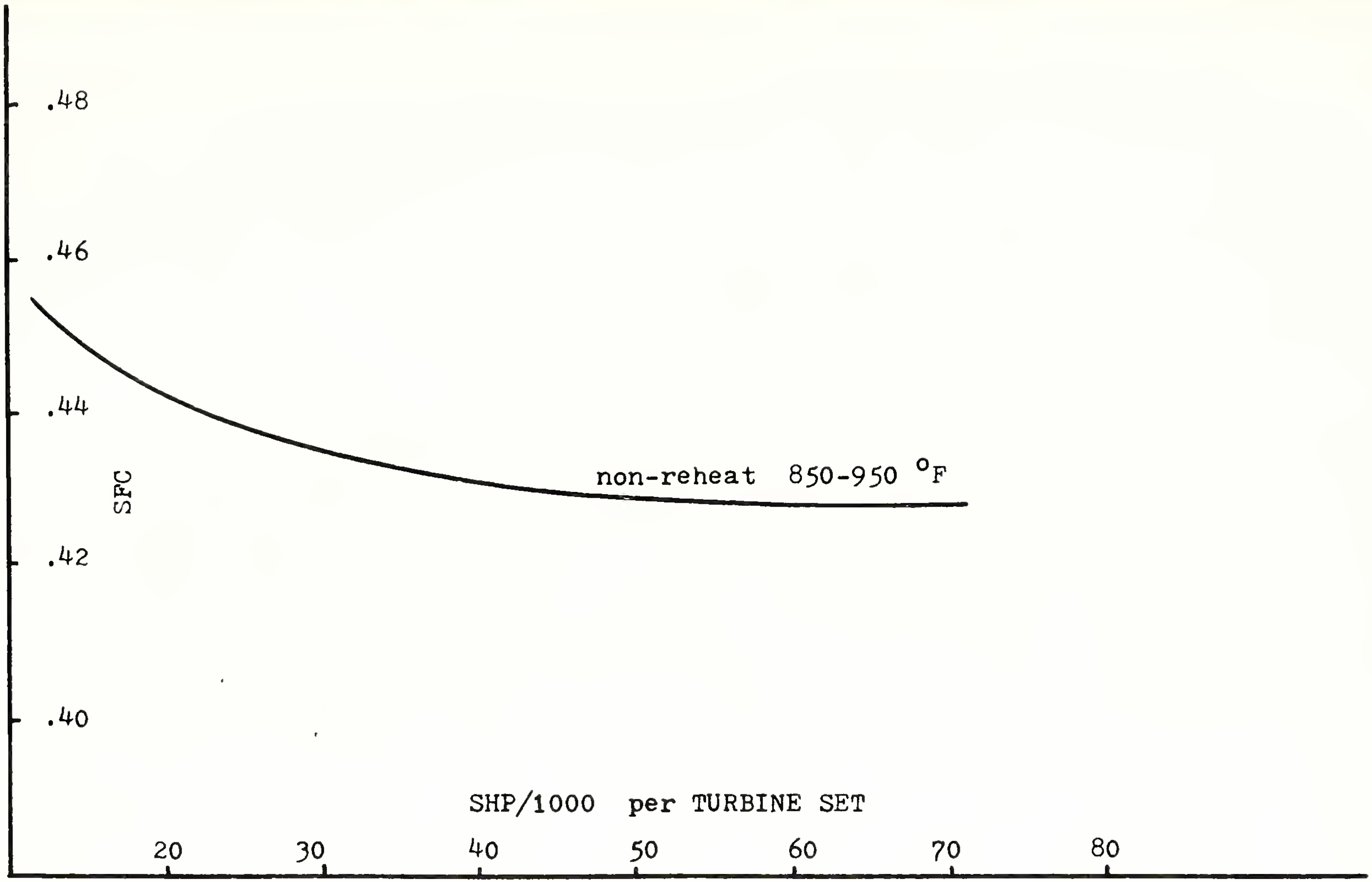


FIGURE V-1 1200 PSI Steam Plant Fuel Rate vs Horsepower

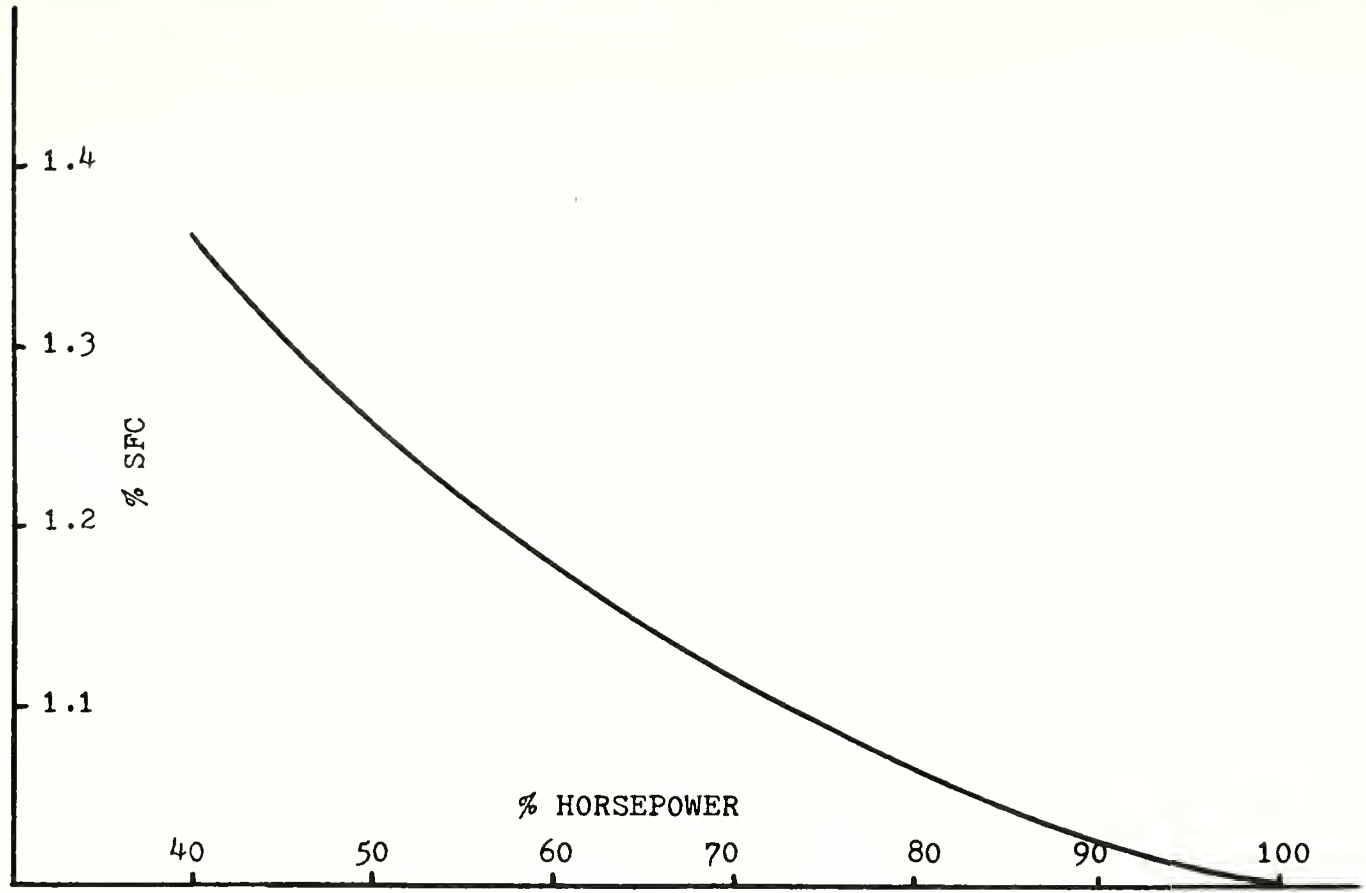


FIGURE V-2 Simple Cycle Gas Turbine Part Load Characteristics (8)

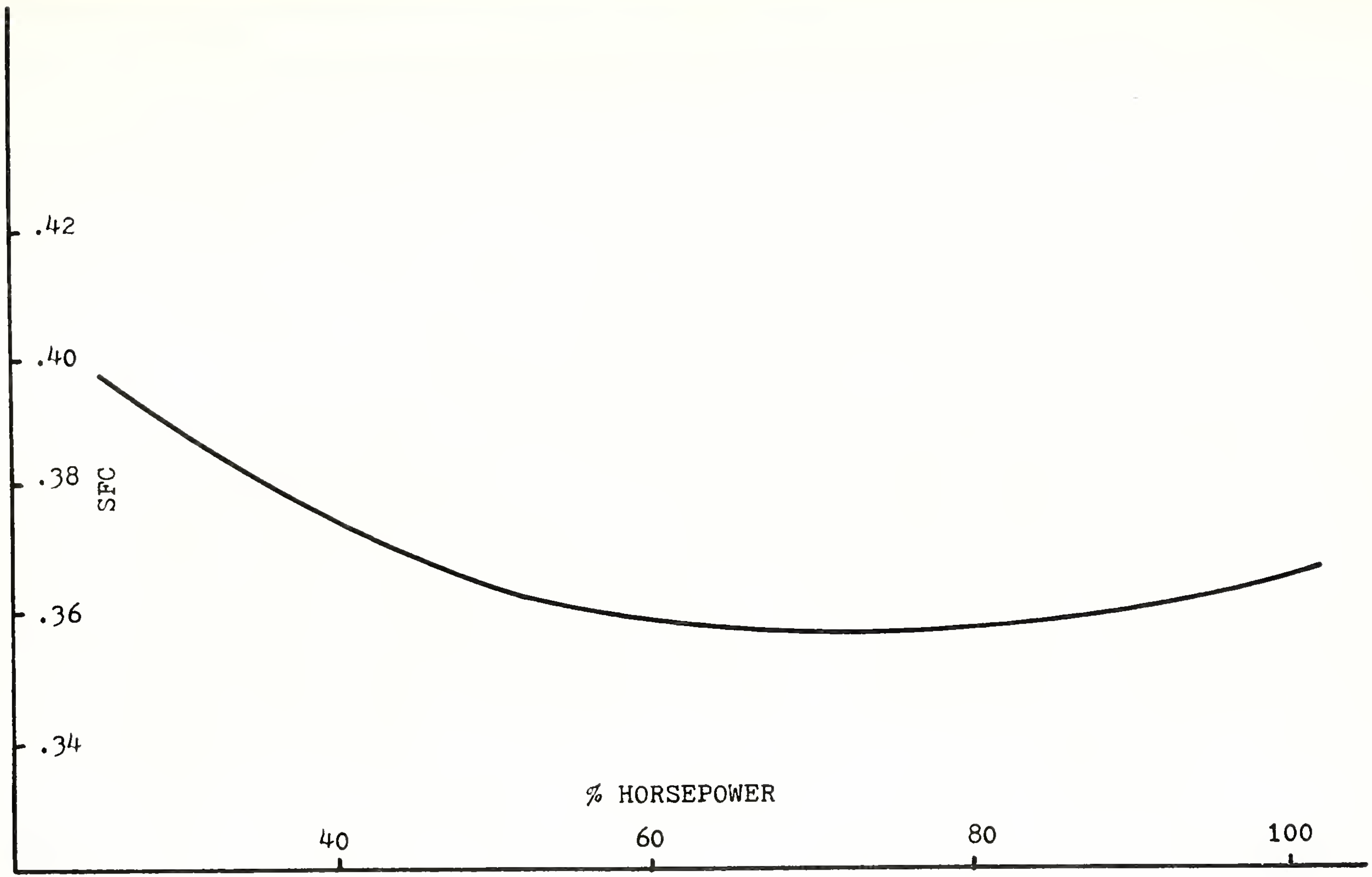


FIGURE V-3 Enterprise Diesels: Fuel Rate vs Per-Cent Horsepower

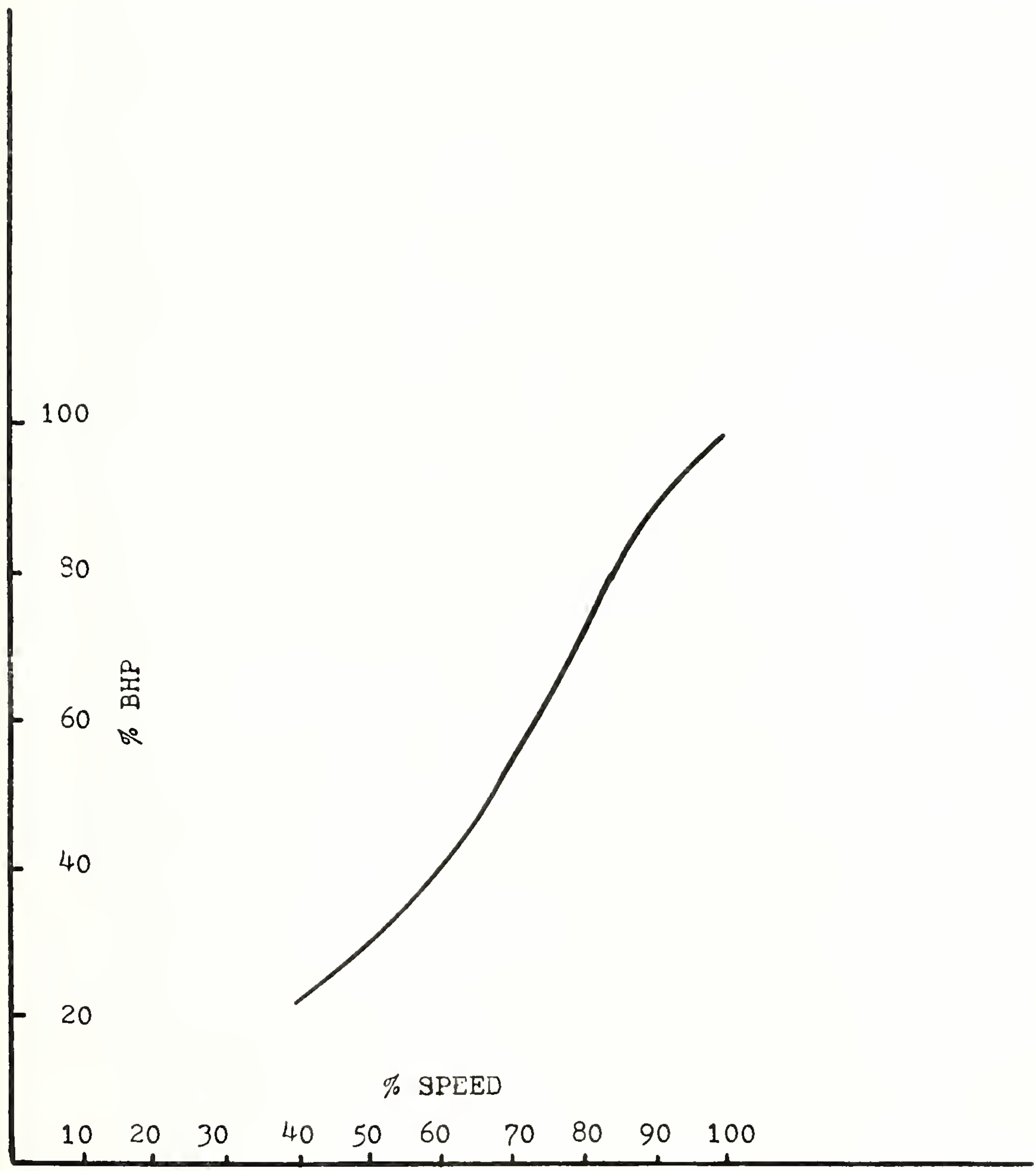


FIGURE V-4 Enterprise Diesels: Horsepower vs RPM

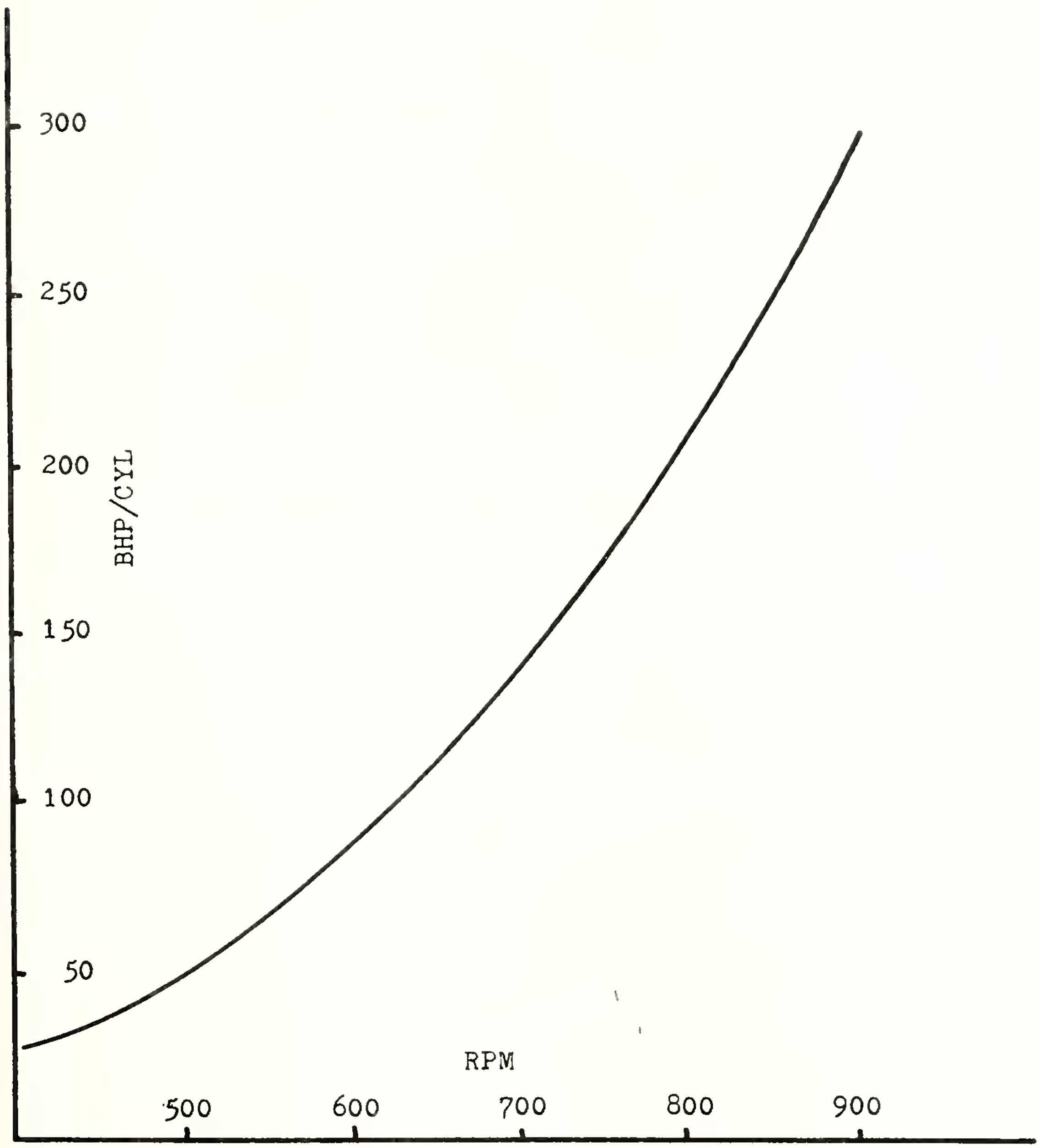


FIGURE V-5 Fairbanks Morse Diesel: Horsepower vs RPM

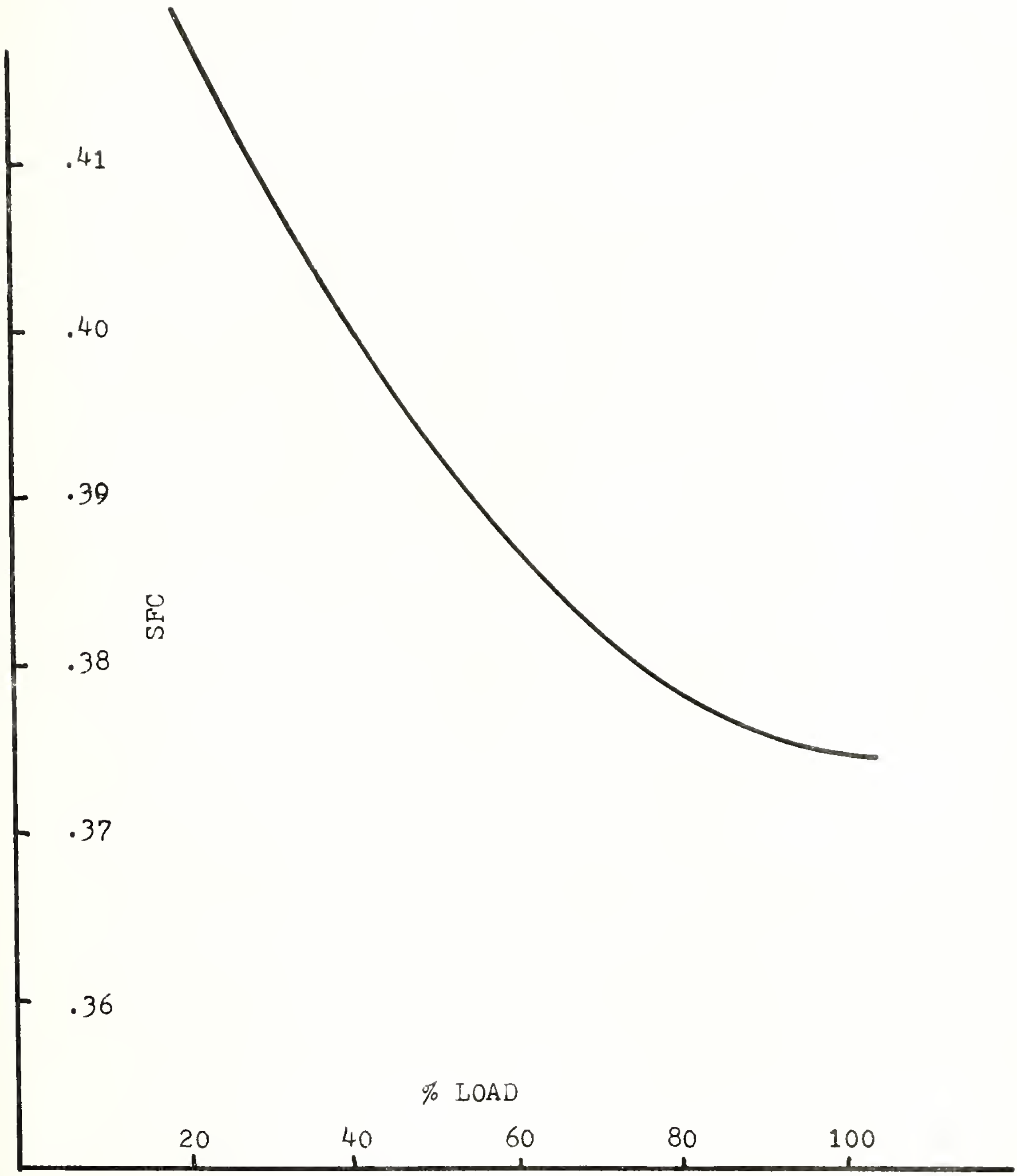


FIGURE V-6 Fairbanks Morse Diesel: Fuel Consumption

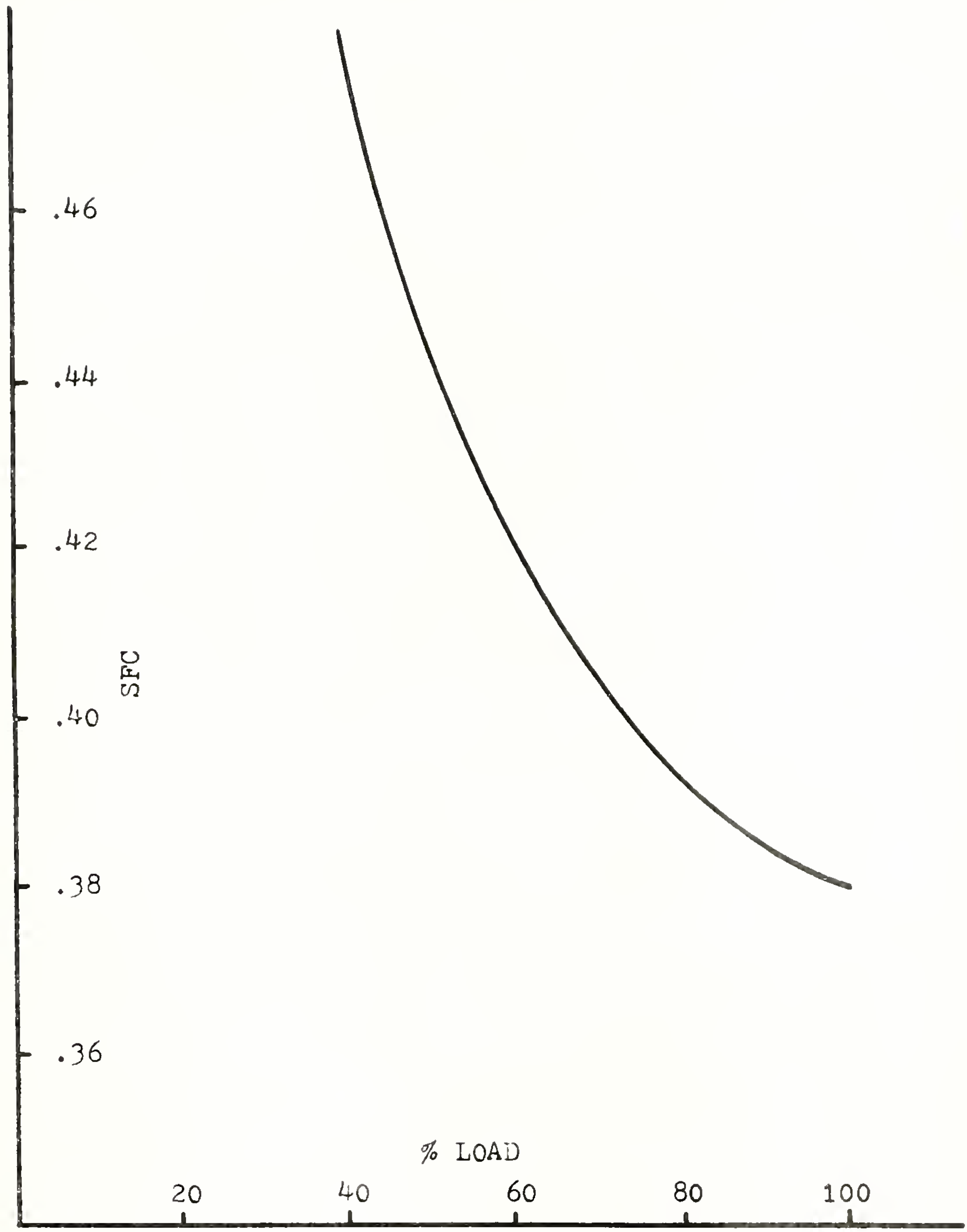


FIGURE V-7 GM-645E5 Diesel: Fuel Consumption

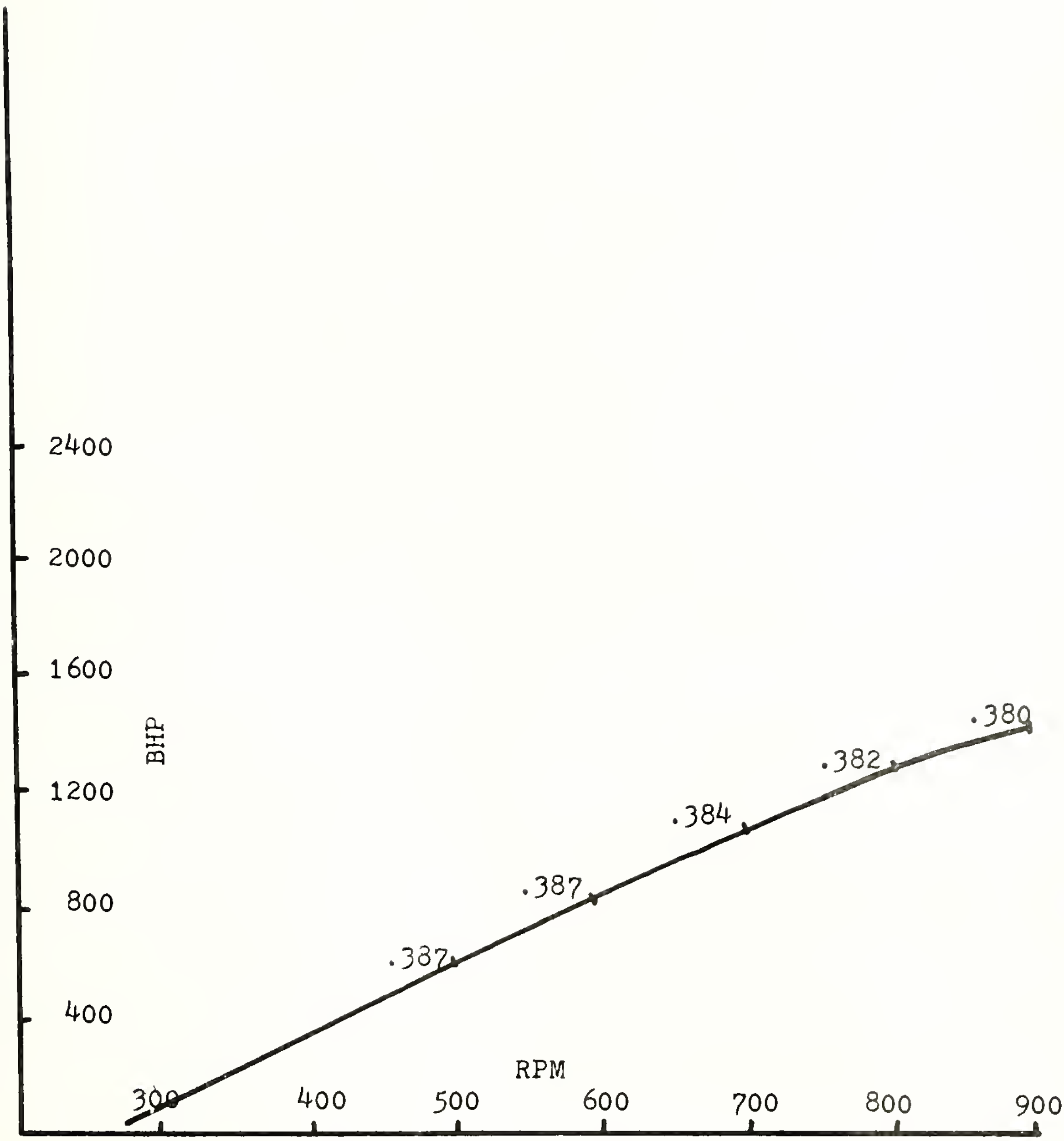


FIGURE V-8 GM-8645E5 Diesel: Horsepower vs RPM

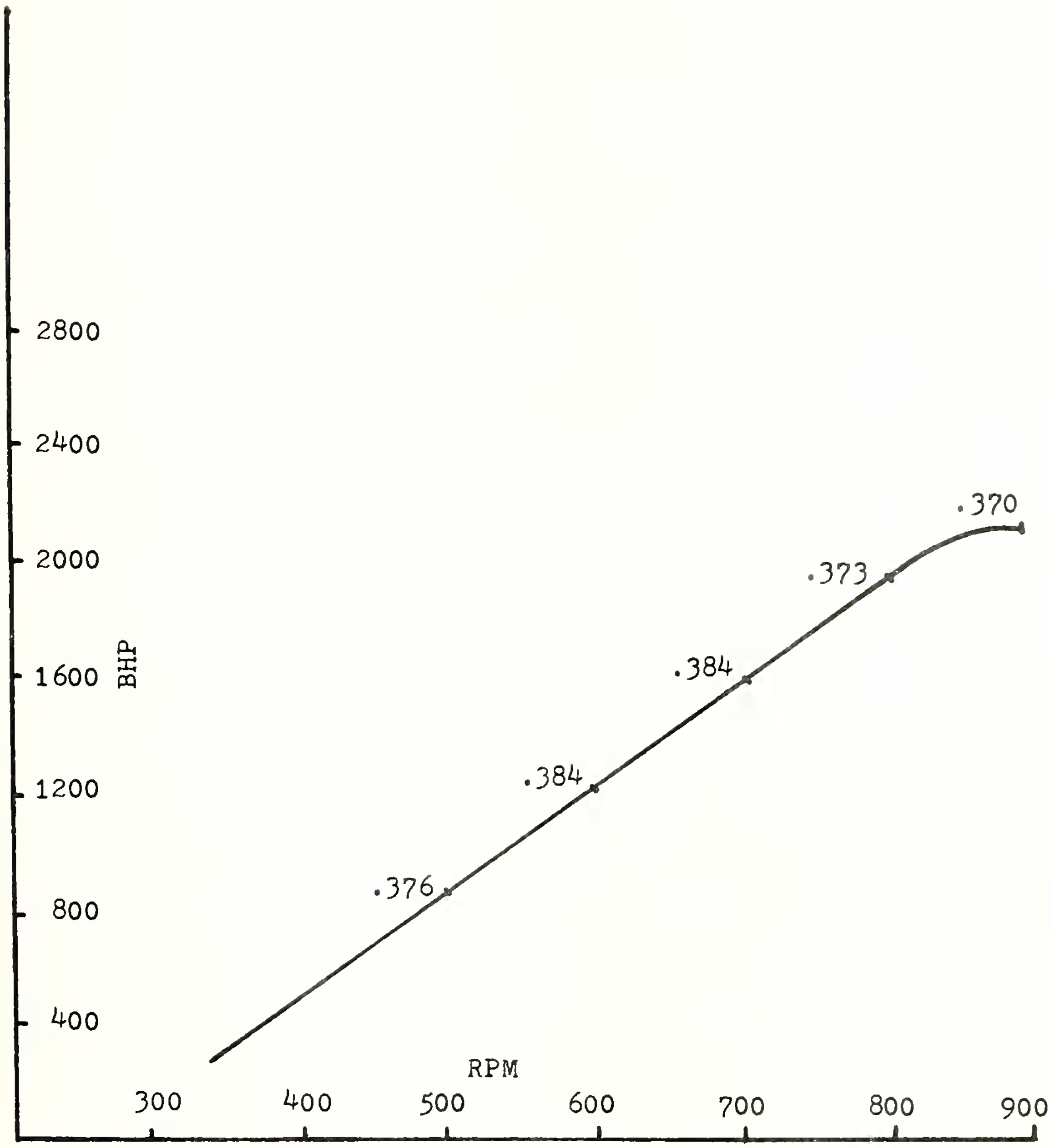


FIGURE V-9 GM 12645E5 Diesel: Horsepower vs RPM

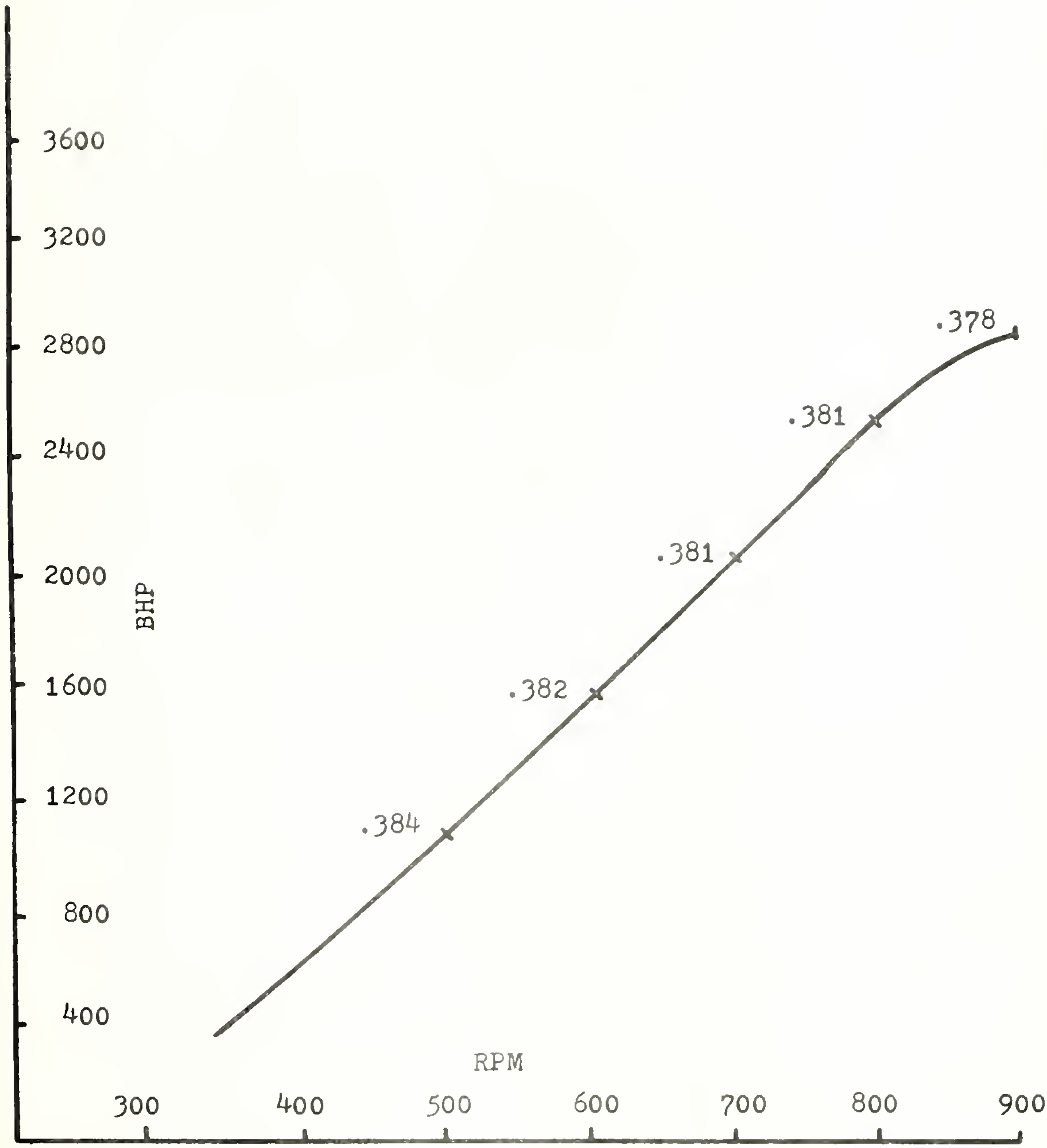


FIGURE V-10 GM-1645E5 Diesel: Horsepower vs RPM

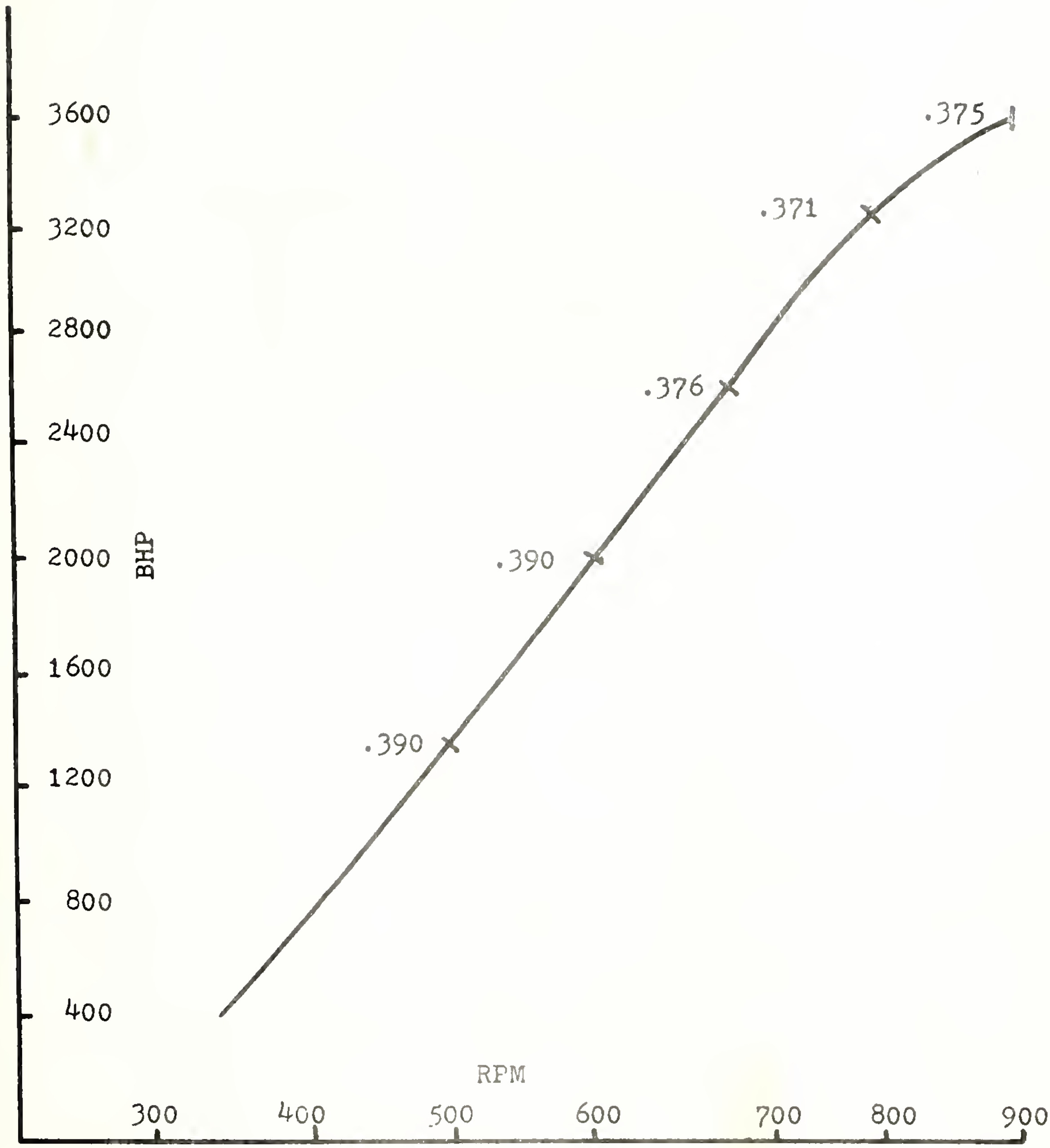


FIGURE V-11 GM-20645E5 Diesel: Horsepower vs RPM

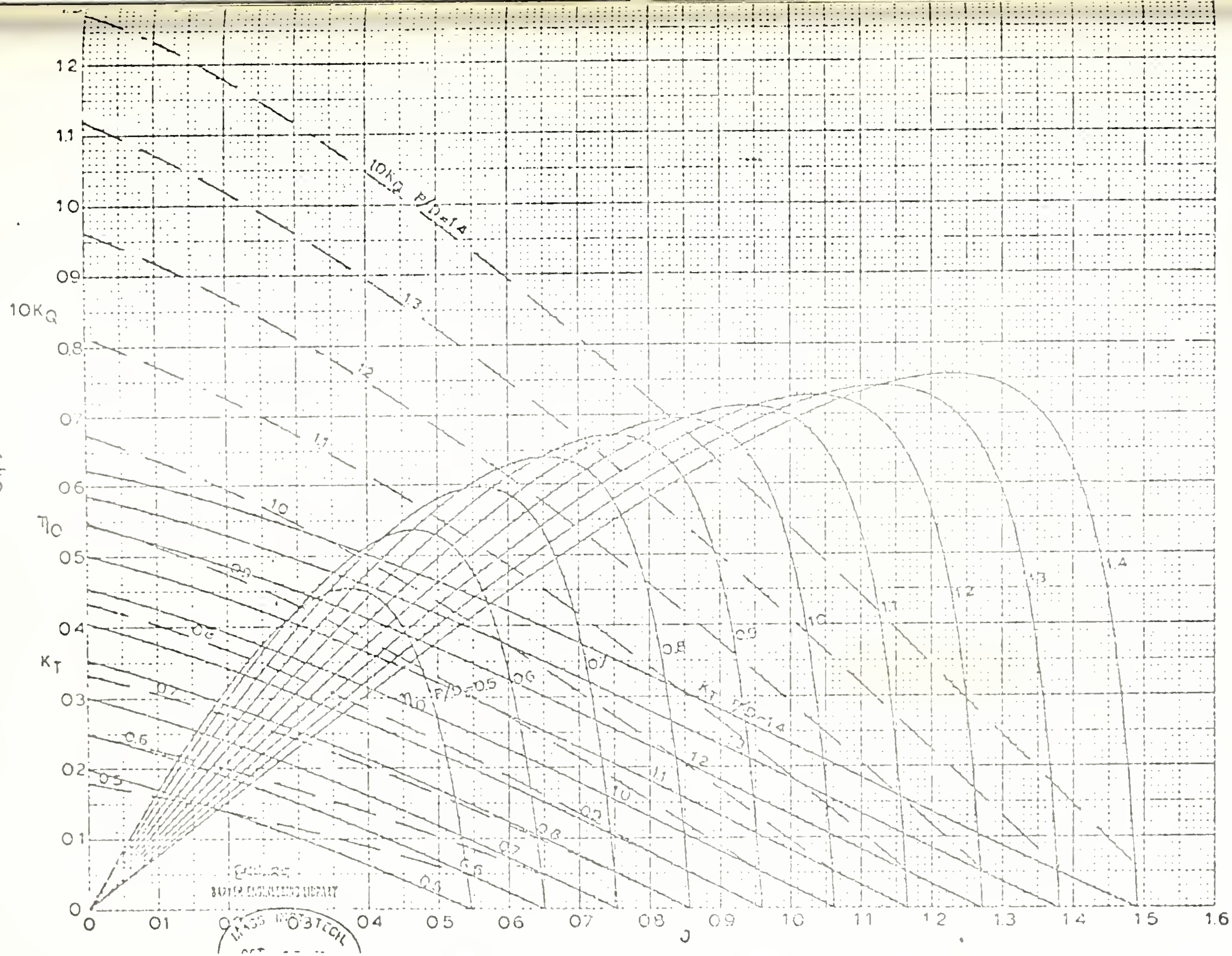


FIGURE V-12 Troost Curve: B4-70

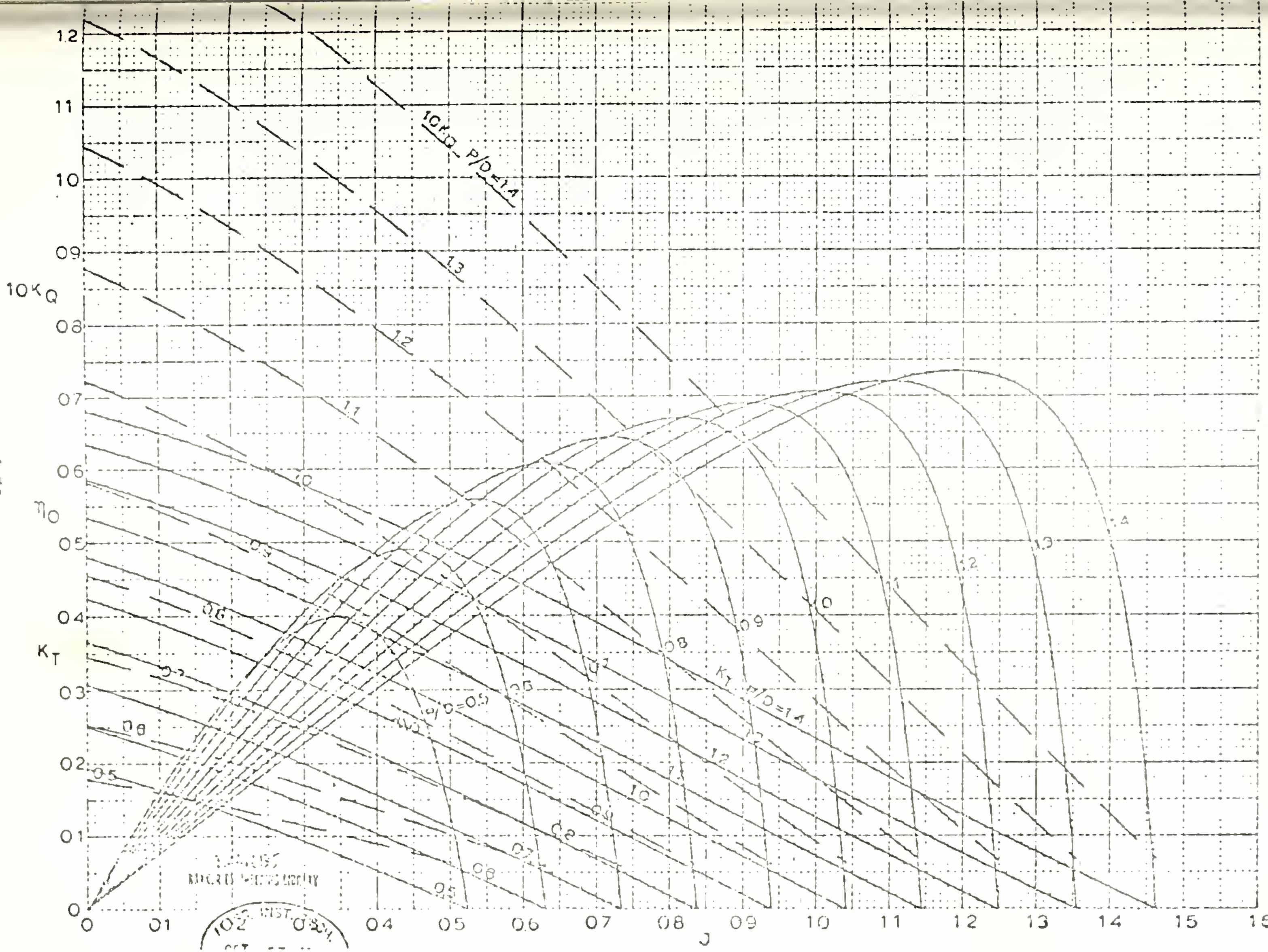


FIGURE V-13 Troost Curve: B4-85

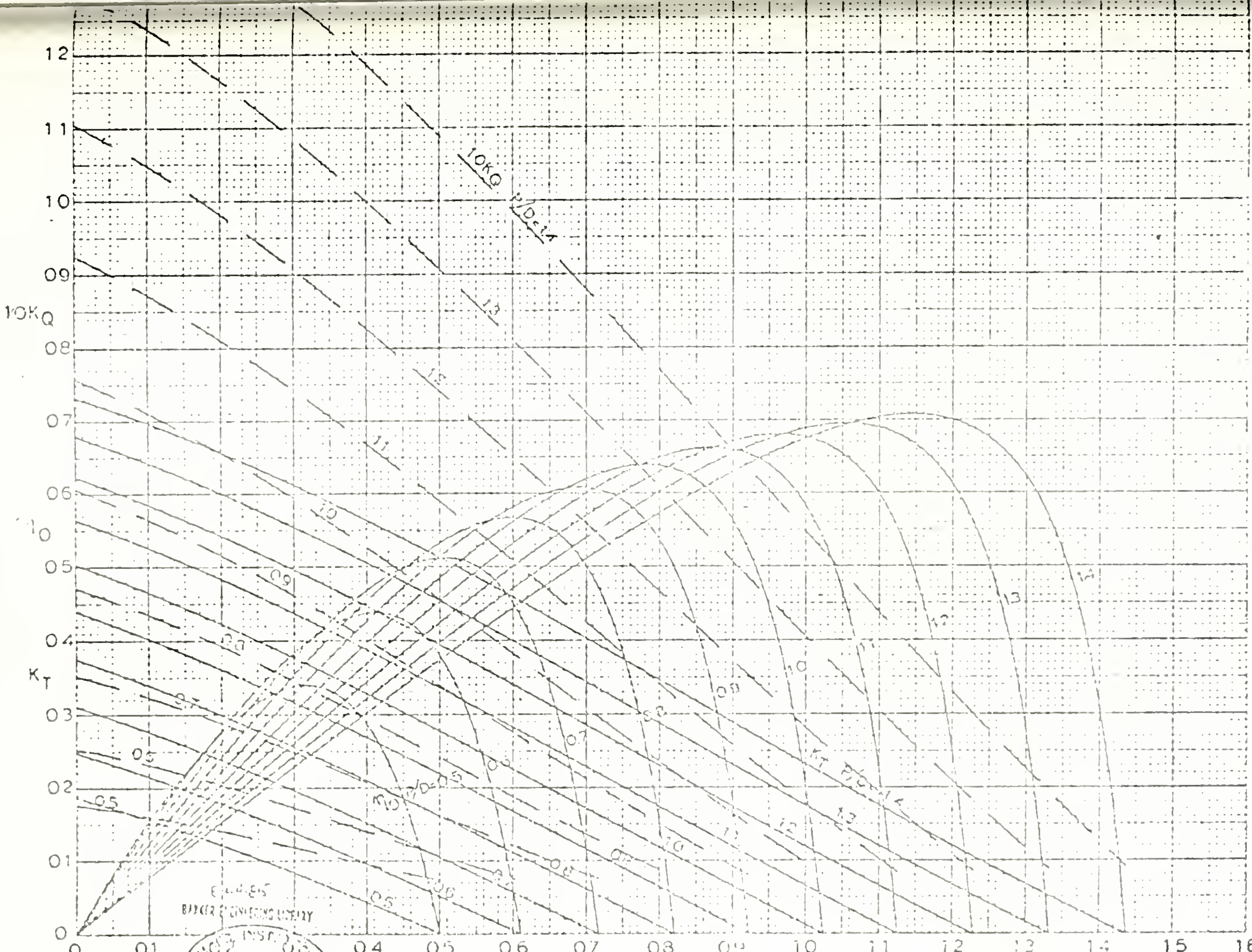


FIGURE V-14 Troost Curve: B4-100

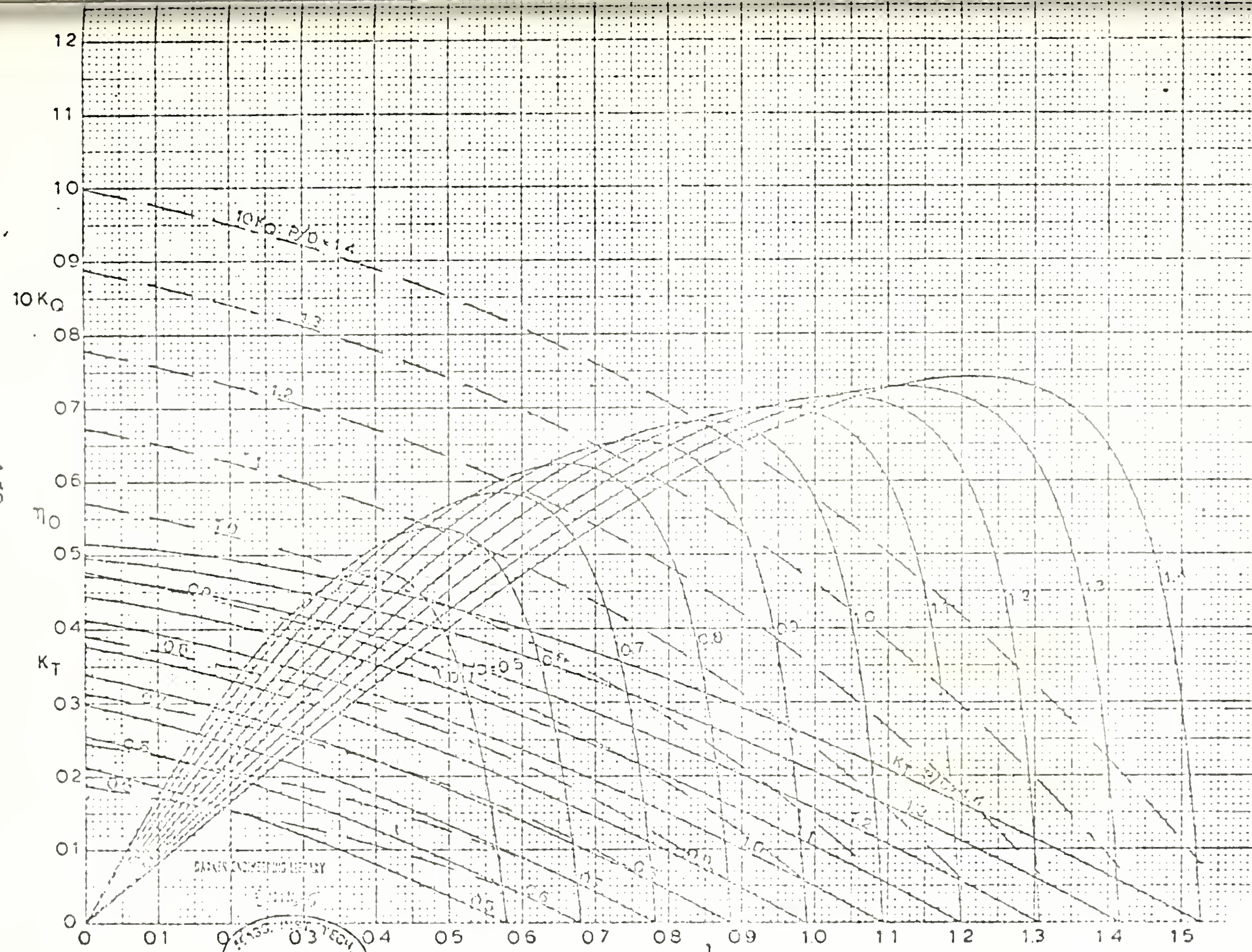


FIGURE V-15 Troost Curve: B5-45

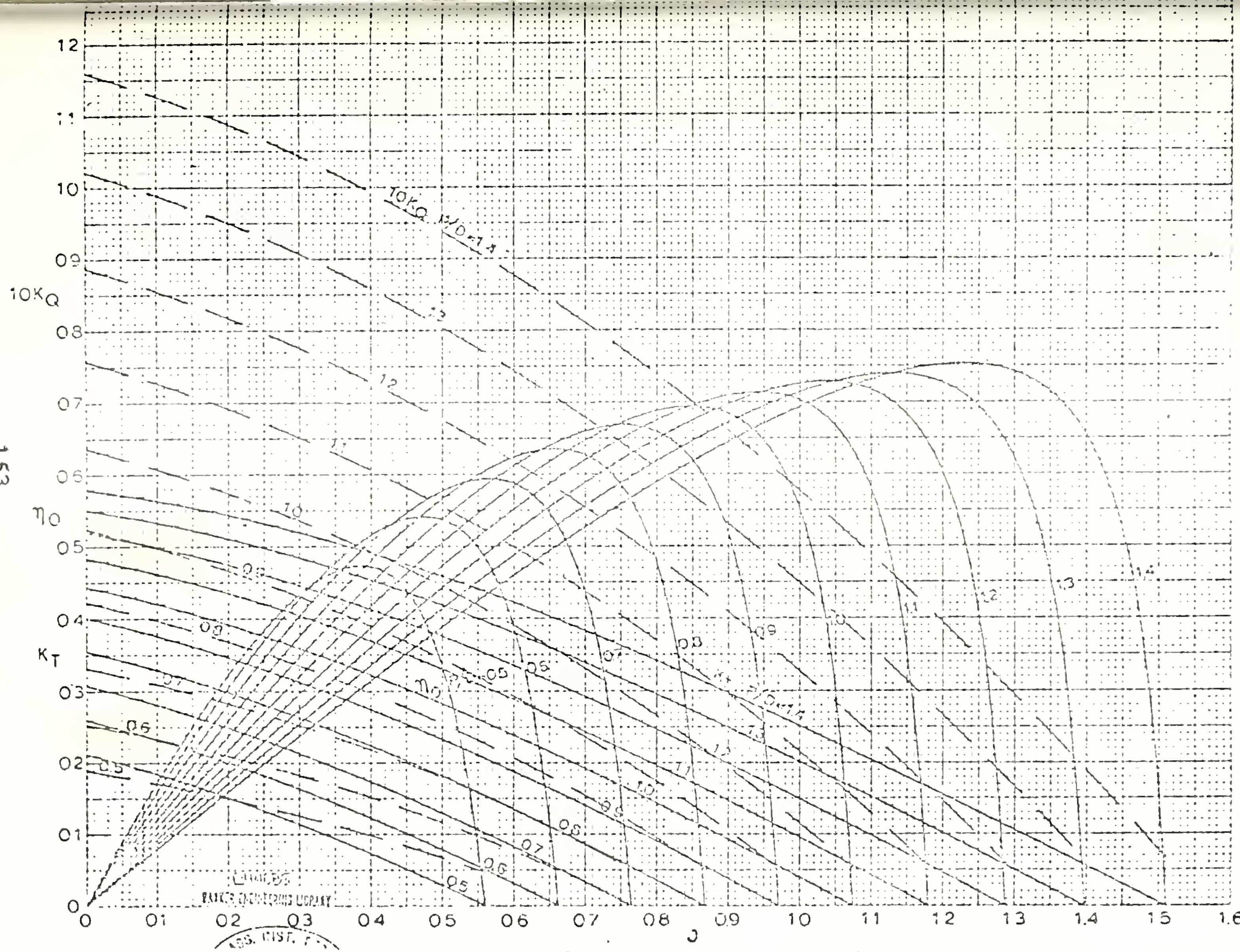


FIGURE V-16 Troost Curve: B5-60

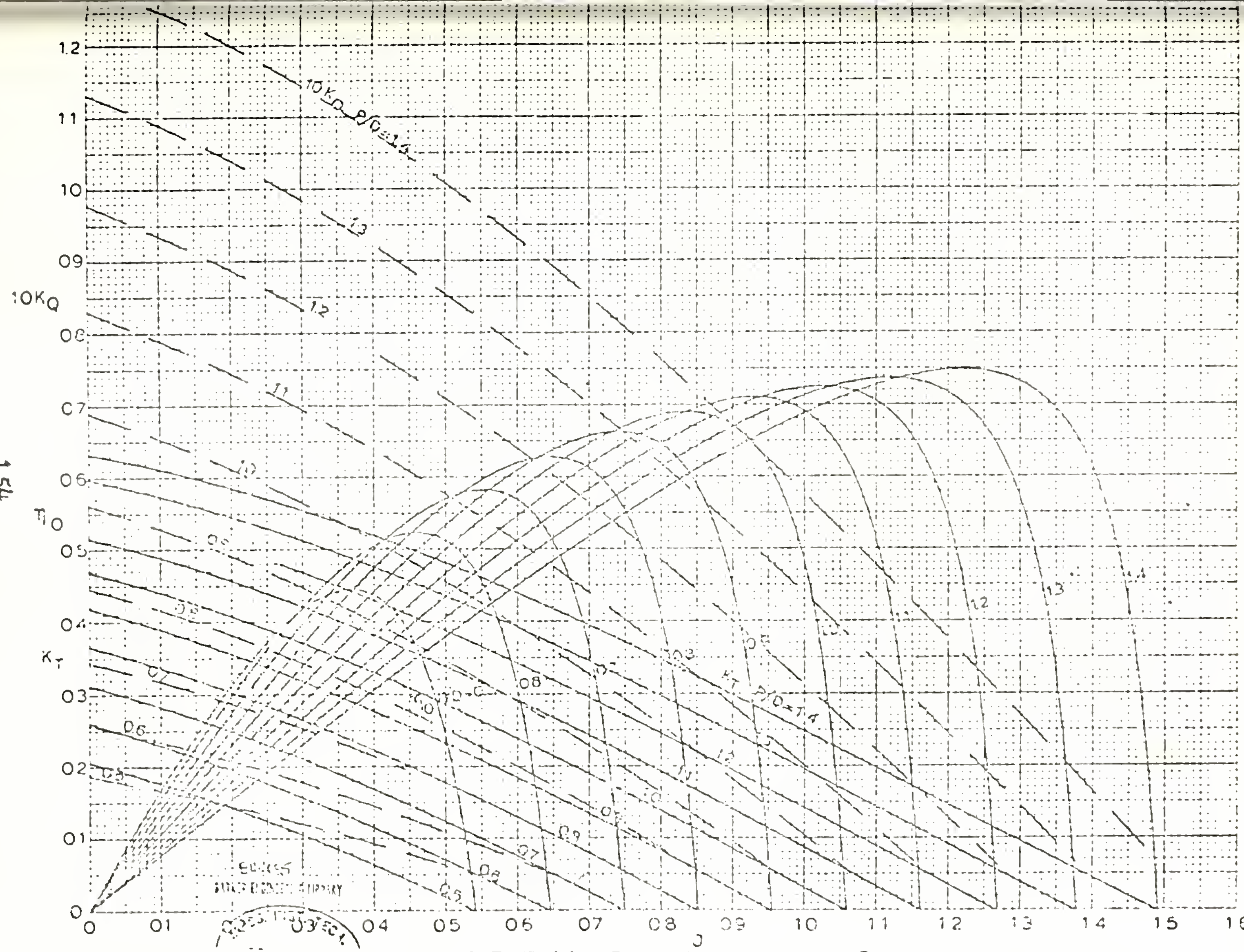


FIGURE V-17 Troost Curve: B5-80

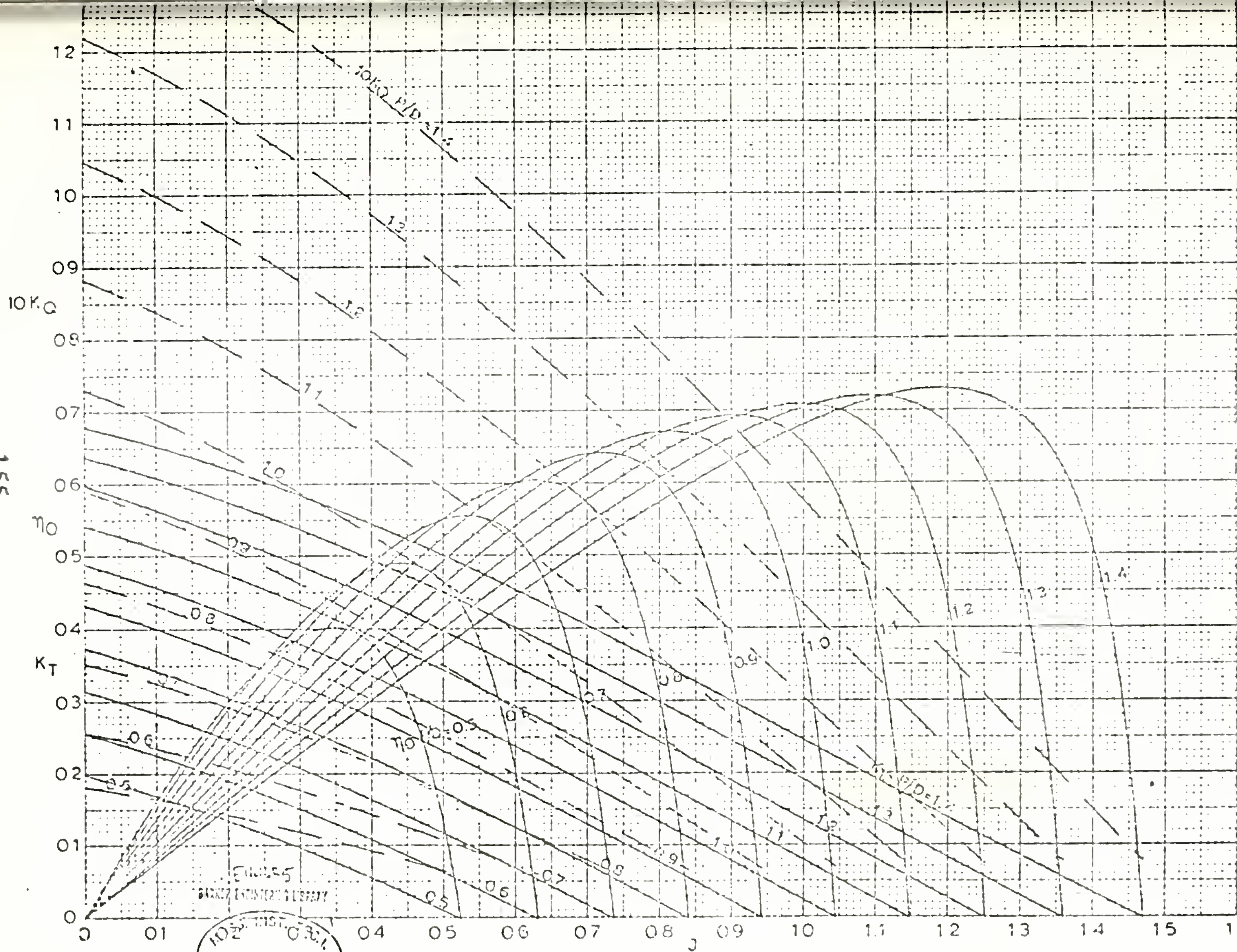


FIGURE V-18 Troost Curve: B5-90

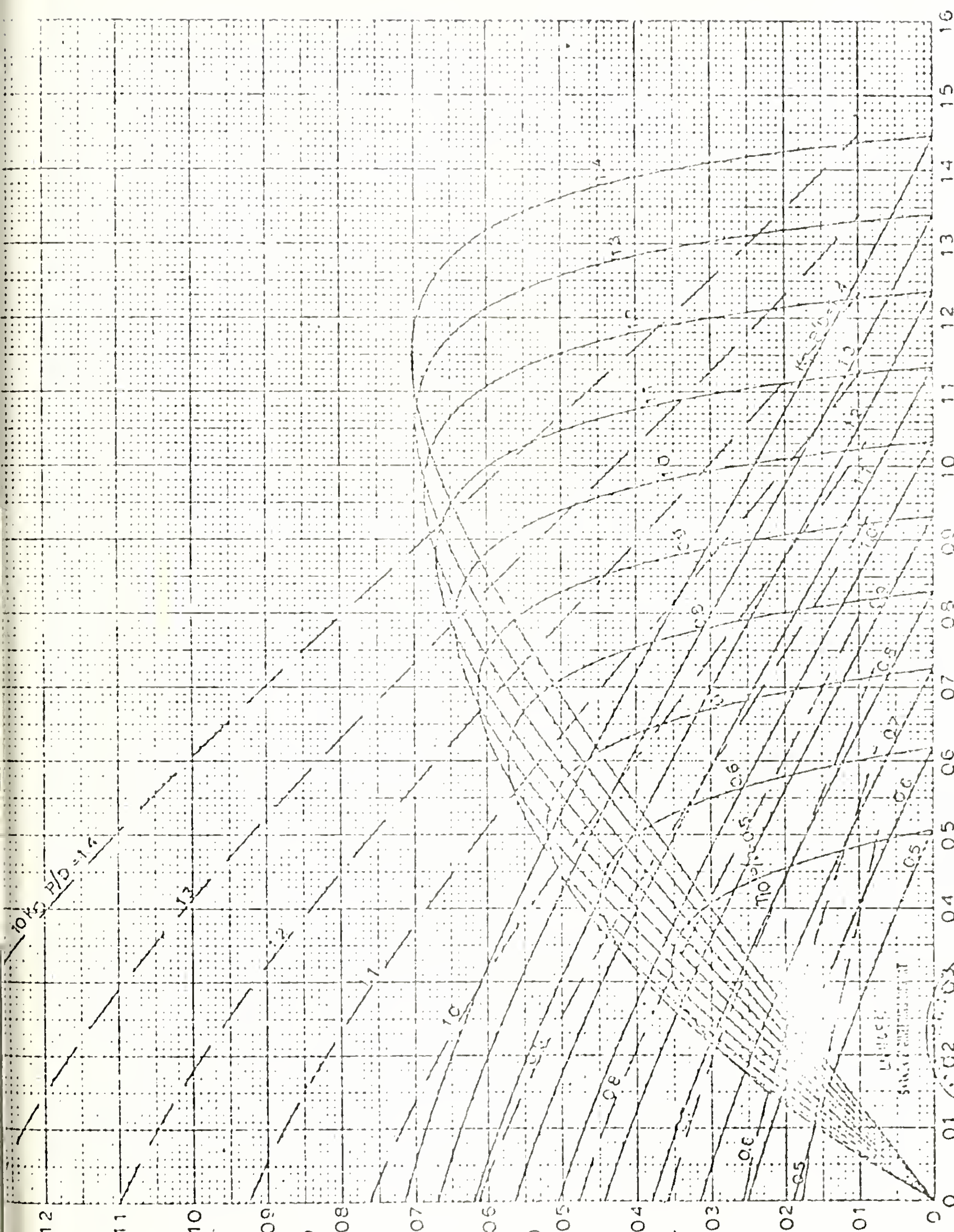


FIGURE V-19 Troost Curve: B5-105

APPENDIX VI - FOM SELECTION ANALYSIS

VI.1 <u>KEY AREAS</u> All propulsion plant designs have preferred characteristics. These key areas are usually singled out by the design requirements, design constraints and design philosophy. For a modern light weight frigate these areas might be as follows:

Cost	Manning
Weight	RMA
Volume	Irradiated Noise
Technical Risk	

VI.2 <u>FOM ASSIGNMENT</u> Assigning an FOM to each of the above areas is not nearly as subjective as presented in the following steps.

STEP 1 Prioritize the key areas

1. Cost
2. Weight
3. Volume
4. Manning
5. RMA
6. Technical Risk
7. Irradiated Noise

STEP 2 Determine the relative weighting of each area by assigning a value of '1' to the most important one, and a lesser value to each of the others. The relative weighting should reflect the importance of each area. This is the part of the analysis that is most easily compromised.

VALUE		REMARKS
Cost.........	1.00	Most important design factor
Weight.......	0.90	Critical factor, but not most important
Volume.......	0.87	Almost as important as weight
Manning......	0.75	An important but not critical area
RMA..........	0.68	About as important in its relationship to manning as weight is to cost
Technical.... Risk	0.55	About half as important as cost in candidate selection
Irradiated... Noise	0.50	Least important of key areas

Assume that prior to this FOM analysis three different propulsion plants were studied and the following results were generated.

	COGOG	CODOG	STEAM
Cost	34.2M	32.6M	31.8M
Weight	24LT	52.8LT	214LT
Volume	2,105FT^3	3,066FT^3	16,000FT^3
Manning (watchstanders)	11	15	34

STEP 3 The propulsion plant design output values (in each of the key areas) are compared and normalized: To determine their relative ratings. The 'best' plant in each area is assigned a value of '10', and the others are rated accordingly.

1. Costs:

Steam			10
CODOG	10	$\frac{31.8}{32.6}$ =	9.7
COGOG	10	$\frac{31.8}{34.2}$ =	9.2

2. Weight:

COGOG			10
CODOG	10	$\frac{24}{52}$ =	4.6
Steam	10	$\frac{24}{214}$ =	1.1

3. Volume:

COGOG			10
CODOG	10	$\frac{2105}{3066}$ =	6.9
Steam	10	$\frac{2105}{16000}$ =	1.3

4. Manning:

COGOG			10
CODOG	10	$\frac{11}{15}$ =	7.3
Steam	10	$\frac{11}{34}$ =	3.2

In areas that are not easily quantified, relative ratings assigned are based on qualatative studies. For this example the following values are used.

5. RMA:

Steam	10
COGOG	7
CODOG	6.5

6. Technical Risk:

Steam	10
COGOG	5
CODOG	5.5

7. Irradiated Noise:

Steam	10
COGOG	7
CODOG	5

STEP 4 Set up a table of the FOM's and the plant ratings. The product of these values represent individual plant FOM's. These plant FOM's are then summed to determine the plant with the highest overall rating.

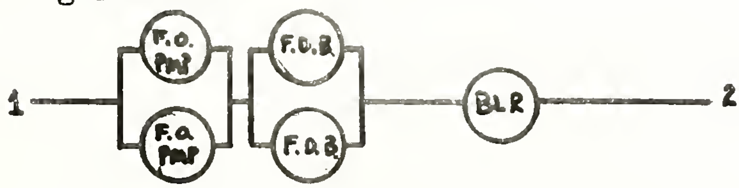
		STEAM		COGOG		CODOG	
KEY AREA	FOM	RATING	FOM	RATING	FOM	RATING	FOM
Cost	1.0	10	10	9.2	9.2	9.7	9.7
Weight	.90	1.1	.99	10	9.0	4.6	4.14
Volume	.87	1.3	1.13	10	8.7	6.9	6.0
Manning	.75	3.2	2.4	10	7.5	7.3	5.48
RMA	.68	10	6.8	7	4.76	6.5	4.42
Tech Risk	.55	10	5.5	5	2.75	5.5	3.03
Irr Noise	.50	10	5	7	3.5	5	2.5
Σ FOM			31.82		45.41		35.27

TABLI VI-1

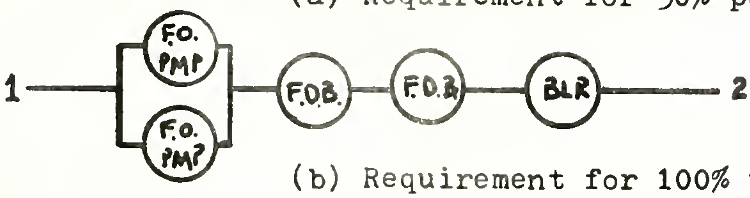
Table VI-1 shows that for the FOM's assigned and the plant ratings, the COGOG plant has a definite advantage over the other two.

APPEDIX VII - RMA

VII.1 FUNCTIONAL SCHEMATIC System and subsystem reliability models consist of a functional schematic. An example of a reliability model for the combustion side of a boiler is shown in Figure VII-1.



(a) Requirement for 50% power



(b) Requirement for 100% power

FIGURE VII-1

For the model in the figure to be operational, a complete 'operational path' must exist between points 1 and 2. Note that in both a and b only one fuel oil service pump is required, yet for 100% power both forced draft blowers must be operational.

An analysis of Figure VII-1 would determine its reliability and availability at both power levels. The designer could then evaluate the effects of variations in the system.

VII.2 <u>RELIABILITY</u> For non-repairable systems and components the following equation defines reliability: $R(t)$ is the system/component reliability.

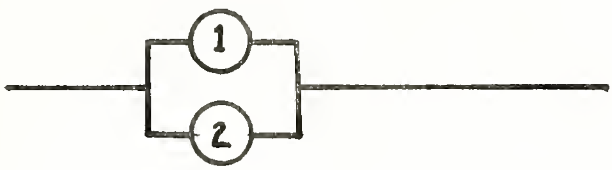
$$R(t) = \exp(t/MTBF) = \text{probability of success} \approx 1-\lambda t \quad (12)$$

$$\lambda = \frac{1}{MTBF}$$

Series components: all components must function properly to obtain the desired results.

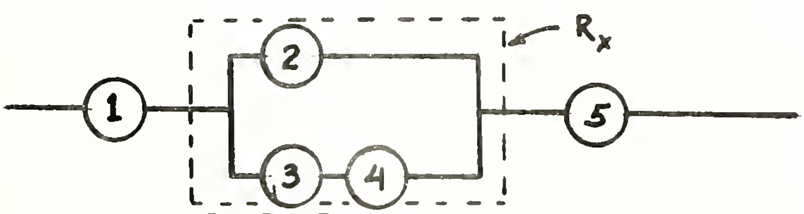
$$R_{sys} = R_1 \cdot R_2 \cdot R_3 \cdot\cdot\cdot\cdot\ R_n$$

Parallel components: either component functioning will provide the desired result.



$$R_{sys} = R_1 + R_2 - R_1 \cdot R_2$$

Series parallel system



$$R_{sys} = R_1 \cdot R_x \cdot R_5$$
$$R_x = R_2 + R_{3,4} - R_2 \cdot R_{3,4}$$
$$R_{3,4} = R_3 \cdot R_4$$
$$R_{sys} = R_1 \cdot (R_2 + R_3 \cdot R_4 - R_2 \cdot R_3 \cdot R_4) \cdot R_5$$

VII.3 AVAILABILITY Series, parallel and series-parallel availability calculations are the same as they are for reliability, but the defination is different.

$$A(\text{inherent}) = \frac{\text{MTBF}}{\text{MTBF+MTTR}}$$

VII.4 EXAMPLE Before an example problem is shown, the reliability of a repairable component should be defined:

$$R(t) = \frac{\mu+\lambda e^{-(\lambda+\mu)t}}{\mu+\lambda} \quad ; \mu = \frac{1}{MTTR} \quad , \lambda = \frac{1}{MTBF} \qquad ; (12)$$

Table VII-1 is a list of reliabilities for various components of a gas turbine plant, and are the basis for the reliability calculations. The reliability model will be for a typical gas turbine plant , including the GT fuel oil system and the reduction gear lube oil system. Figure VII-2 shows the individual subsystems as well as the complete system.

Reliability of fuel oil system

$$R_{m,p} = (R_{mtr} \cdot R_{pmp}) + (R_{mtr} \cdot R_{pmp}) - R^2_{mtr} \cdot R^2_{pmp}$$

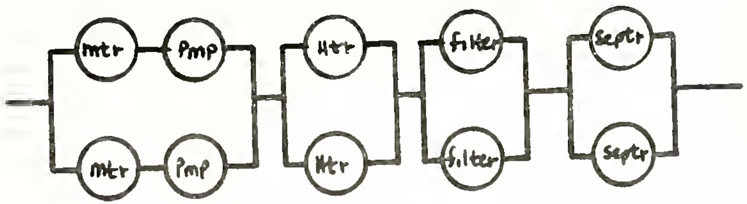
$$R_{m,p} \cong 1.0$$

COMPONENT	MTBF	MTTR	$R=\begin{cases} \frac{\mu+\lambda e^{-(\lambda+\mu)t}}{\mu+\lambda} & : \text{Repairable} \\ 1-\lambda t & : \text{NR (non-Repairable)} \end{cases}$
LM 2500	4,000	24	.994
CLUTCH	50,000	NR	.986
REDUCTION GEAR	200,000	NR	.996
SHAFT & BEARINGS	200,000	NR	.996
CRP PROPELLER	25,000	15	.999
FUEL OIL MOTOR	7,500	18	.998
FUEL OIL PUMP	5,500	4.5	.999
FUEL OIL HEATER	15,000	3.8	1.00
FUEL OIL FILTER	60,000	3.0	1.00
FUEL SEPERATOR	10,000	4.0	1.00
LUBE OIL MOTOR	7,500	7.8	.999
LUBE OIL PUMP	4,000	5.0	.999
LUBE OIL STRAINER	60,000	3.0	1.00
LUBE OIL COOLER	90,576	3.0	1.00
SALT WATER COOLING MOTOR	27,000	9.2	1.00
SALT WATER COOLING PUMP	12,500	7.6	.999
BOILER	15,000	12	.999
FORCED DRAFT BLOWER: MTR	4,200	7	.998
FORCED DRAFT BLOWER: TRB	2,800	6	.997
MAIN FEED PUMP	2,500	14	.994
MAIN FEED BOOSTER PUMP	5,500	14	.997
STEAM PROPULSION TURBINE	100,000	10	1.00
FIXED PITCH PROPELLER	200,000	NR	.996
MAIN CONDENSER	50,000	5	1.00
MAIN CONDENSATE PUMP	5,400	10	1.00
MAIN LUBE OIL PUMP	4,000	5	.999
TURBINE GENERATOR	3,500	8	.998
DIESEL GENERATOR	1,900	8	.996
LUBE OIL PURIFIER	10,000	4	1.00

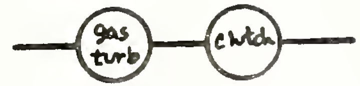
The above reliabilities are based on a 30 day (720 hrs) operational time.

TABLE VII-1 RELIABILITIES OF VARIOUS PROPULSION PLANT COMPONENTS (12)

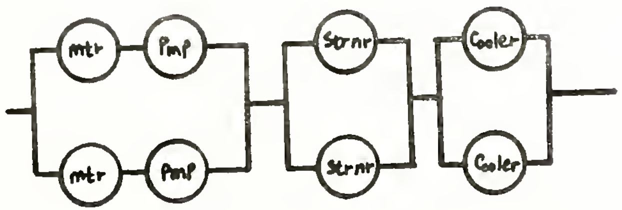
MODEL OF GAS TURBINE PLANT



FUEL OIL SYSTEM



GAS TURBINE



LUBE OIL SYSTEM

TRANSMISSION

FIGURE VII-2 SYSTEM AND SUBSYSTEM FUNCTIONAL SCHEMATICS

$$R_{htr} = 2 \cdot R_{htr} - R^2_{htr}$$

$$R_{htr} \cong 1.0$$

$$R_f = 2 \cdot R_f - R^2_f$$

$$R_f \cong 1.0$$

$$R_s = 2 \cdot R_s - R^2_s$$

$$R_s \cong 1.0$$

$$R_{f.o.\ sys} = R_{m,p} \cdot R_{htr} \cdot R_f \cdot R_s$$

$$R_{f.o.\ sys} \cong 1.0$$

Reliability of lube oil system

$$R_{m,p} = (R_{mtr} \cdot R_{pmp}) + (R_{mtr} \cdot R_{pmp}) - R^2_{mtr} \cdot R^2_{pmp}$$

$$R_{m,p} \cong 1.0$$

$$R_{strnr} = 2 \cdot R_{strnr} - R^2_{strnr}$$

$$R_{strnr} \cong 1.0$$

$$R_{clr} = 2 \cdot R_{clr} - R^2_{clr}$$

$$R_{clr} \cong 1.0$$

$$R_{l.o.\ sys} = R_{m,p} \cdot R_{strnr} \cdot R_{clr}$$

$$R_{l.o.\ sys} \cong 1.0$$

Reliability of gas turbine system

$R_{gt} = .994$

$R_{cl} = .986$

$R_{gt\ sys} = R_{gt} \cdot R_{cl}$

$R_{gt\ sys} = .98$

Reliability of reduction gear, shafting & bearings and crp propeller

$R_{rg} = .996$

$R_{sh\&brg} = .996$

$R_{crp} = .999$

$R_{trans} = R_{rg} \cdot R_{sh\&brg} \cdot R_{crp}$

$R_{trans} = .991$

Reliability of entire gas turbine system

$R_{gt\ sys} = R_{f.o.\ sys} \cdot R_{l.o.\ sys} \cdot R_{gt\ sys} \cdot R_{trans}$

$R_{gt\ sys} = .971$

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